IRON AND STEEL ENGINEER

J. F. KELLY, Editor

CONTENTS

Volume I MAY, 1924 Number 5

Iron and Steel Exposition 202
Program of Meetings, Papers, Etc. 204
Editorials 205
Application of Taper Roller Bearings to Motors in the Steel Industry—By T. V. Buckwalter 210
Application of Anti-Friction Bearings to Motors in the Steel Industry—By F. W. Gurney 214
Mounting and Maintenance of Roller Bearings on Electric Motors—By D. E. Batesole 217
Ball Bearing Application and Maintenance—By H. R. Reynolds 221
Practical Application of Anti-Friction Bearings to Electric Motors—By H. E. Brunner 225
Selection, Installation and Operation of Anti-Friction Bearings in Heavy Duty Service—By J. T. R. Bell 229
Mounting and Maintenance of Roller Bearings in Electric Motors—By W. B. Wachtler 234
Design and Application of Ball Bearings to Motors—By T. D. Crow 245
Discussion on Ball and Roller Bearings (Including all discussion presented on preceding papers) 256
The Magnetized Roller Conveyor—By Harry Hollis 248
Third Rail Locomotive for Yard Service—By F. O. Schnure 250
Load Regulator for Use in Connection with Turbo-Generators—By F. O. Schnure. 251
Automatic Electric Stations for Steel Mills—By Chester Lichtenberg 252

(List of Advertisers Page 208)

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APPLICATION FOR SPACE

ANNUAL EXHIBIT OF

ELECTRICAL, MECHANICAL, COMBUSTION, METALLURGICAL APPARATUS

Under Auspices of the

IRON AND STEEL ENGINEERS
A. I. & S. E. E.

DUQUESNE GARDEN **PITTSBURGH, PA.**

September 15, 16, 17, 18, 19 and 20, 1924

The form below should be used in making application for space.

In the interest of the individual exhibitor, the exhibit as a whole, the Iron and Steel Industry, the exhibits will be classified and assignments made after the management has full knowledge of the character of proposed exhibits and exhibitors' requirements. Each applicant is requested to indicate his preference of location, after which consistent assignments will be made accordingly.

Check for $25.00 must accompany application for space, which will be credited as part payment of booth. In case of cancellation this fee is not refunded.

Iron and Steel Exposition,
706 Empire Building,
Pittsburgh, Pa.

Dear Sirs:

In accordance with and subject to the Rules and Regulations governing this Exhibit, the undersigned hereby makes application for space at the Exhibit to be held in conjunction with the Annual Convention of the Association of Iron and Steel Electrical Engineers at Duquesne Garden, Pittsburgh, Pa., September 15th to 20th, 1924.

Our preferences of location as shown on the official floor plan are as follows:

First Choice..... Second Choice..... Third Choice..... Fourth Choice.....

The following is a general statement of the character of our proposed exhibit:

.....

.....

.....

.....

We will require electric power for.....

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We will require compressed air and gas for.....

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Firm .....

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Date .....

Check in the amount of $25.00 is enclosed.

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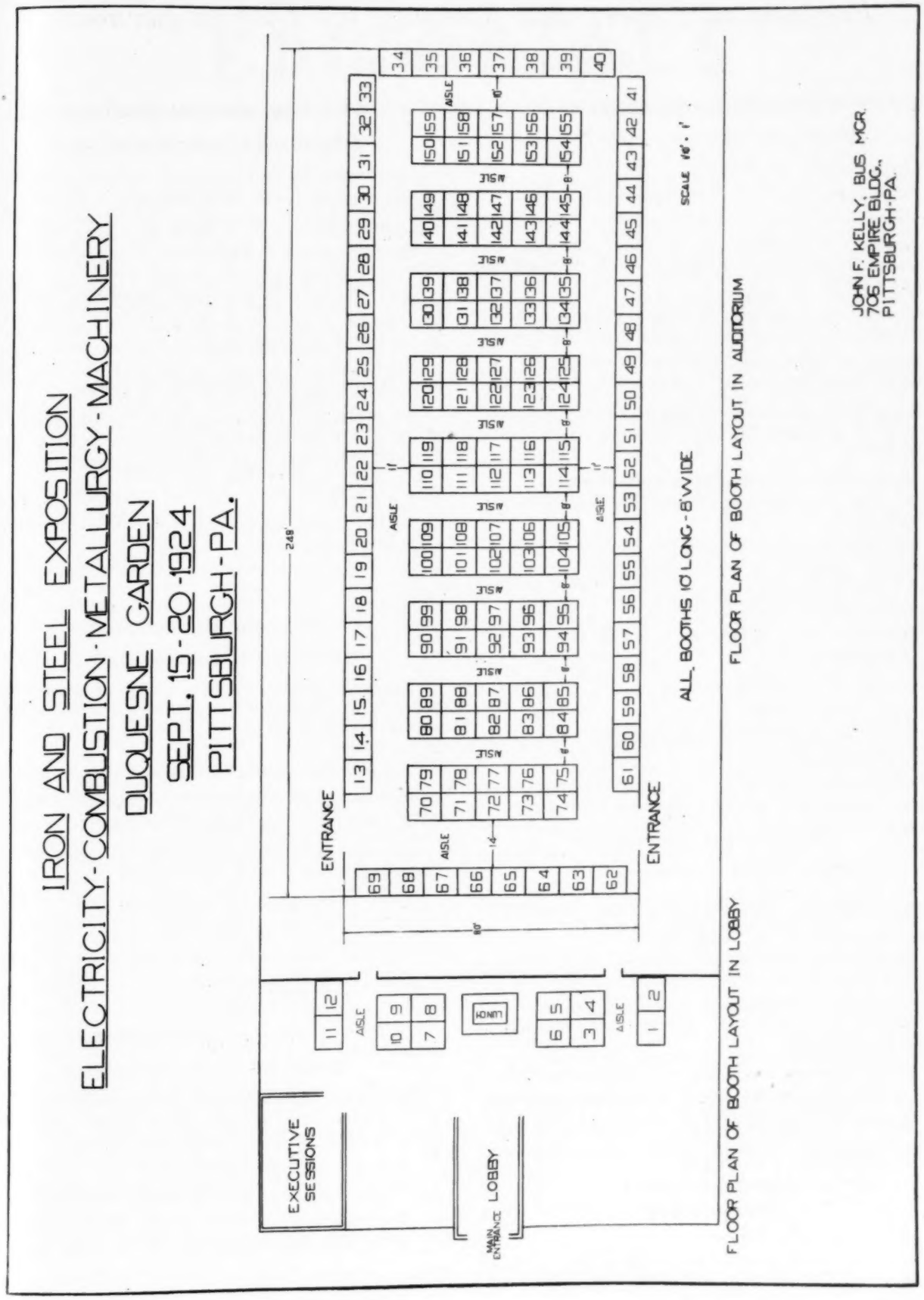
IRON AND STEEL EXPOSITION
ELECTRICITY-COMBUSTION-METALLURGY-MACHINERY
DUQUESNE GARDEN
SEPT. 15-20-1924
PITTSBURGH-PA.
EXECUTIVE SESSIONS
MAIN ENTRANCE
LOBBY
LUNCH
ENTRANCE
ENTRANCE
AISLE
ALL BOOTHS 10' LONG-8' WIDE
SCALE 16' = 1'
FLOOR PLAN OF BOOTH LAYOUT IN LOBBY
FLOOR PLAN OF BOOTH LAYOUT IN AUDITORIUM
JOHN F. KELLY, BUS MGR.
705 EMPIRE BLDG.,
PITTSBURGH-PA.

Program of Papers and Meetings for District Sections of the A. I. & S. E. E.

Philadelphia Section

Officers and Committees—1923-24

Chairman—W. H. Burr.

Past Chairman—W. C. Kennedy.

Vice Chairman—P. T. Vanderwaart.

Executive Committee—W. H. Burr, chairman; P. T. Vanderwaart, W. C. Kennedy, F. H. Woodhull, George Pfeffer.

Papers Committee—H. S. Boone, chairman; A. J. Standing, J. R. Penman, F. O. Schnure, J. R. Bower.

Meetings Committee—B. T. Hare, chairman; T. E. Gallagher, C. H. Williams, C. J. Maloney, A. B. Coffman.

Reception Committee—George H. Schaeffer, chairman; L. R. Humpton, H. D. Esterly, H. G. Nichols, E. D. Sibley.

Membership and Attendance Committee — D. M. Petty, chairman James E. Wilson, H. N. Lucas, R. F. Gale, G. W. C. Frey.

Representatives to the Engineers' Club—Board of Directors, Linn O. Morrow; Meetings Committee, John C. Reed; Publications Committee, David Smith; Research Committee, H. A. Lewis.

Secretary—Linn O. Morrow, Franklin Trust Building, Philadelphia, Pa.

Last Meeting

Saturday, June 7, 1924—

Annual outing. Bring your friends. Complete announcements later.

Pittsburgh Section

Officers and Committees—1923-24

Chairman—J. J. Booth.
Past Chairman—R. H. Keil.
Vice Chairman—R. E. Lewis.
Secretary—J. F. Kelly.
Assistant Secrtary and Treasurer—T. Moffitt.

Executive Committee—J. J. Booth, chairman; R. H. Keil, R. E. Lewis, J. H. Barker, H. C. Mosley, J. F. Kelly, T. Moffitt.

Papers Committee—R. E. Lewis, chairman.

Meeting Committee—W. C. McKibben, chairman.

Membership Committee—R. M. Hussey, chairman.

Entertainment Committee—E. W. Harry, chairman.

Meetings and Papers

Inspection Trip

Saturday, June 21, 1924—

National Tube Company, McKeesport, Pa.; Clairton By-Product Coke Works, Carnegie Steel Company, Clairton, Pa. Announcements later.

Chicago Section

Officers and Committees—1923-24

Chairman—Henry Davis.

Vice Chairman—R. L. McIntosh.

Committeeman—W. S. Hall.

Committeeman—Gordon Fox.

Attendance Committee—C. W. Donovan, chairman; F. A. Wiley, C. J. Smith, V. A. Bloxham, W. D. Cameron, F. L. Collins, A. R. Dibbens, R. E. Boehler, E. C. Otto, F. J. Burd.

Entertainment Committee—B. G. Beck, chairman; R. O. Herbig, T. P. McGinnis, L. M. Gumm.

Reception Committee—Gordon Fox, chairman; O. M. Bercaw, L. M. Gumm, G. H. Lounsbery, E. C. Otto.

Meetings and Papers

May 16, 1924—

An illustrated lecture. Meetings at Atlantic Hotel, Chicago, Ill.

Cleveland Section

Officers and Committees—1923-24

Executive Committee—George R. Roemer, chairman; A. W. Mohrman, vice chairman; A. F. Jones, past chairman; R. E. Scovel, L. W. Long, J. B. Childs, R. S. Sawdey.

Papers Committee—S. L. Henderson, chairman.

Meetings Committee—H. W. Eastwood, chairman.

Membership Committee—C. B. Skelly, chairman.

Secretary—A. R. Lintern, 7960 Lorain Avenue, Cleveland, Ohio.

Meetings and Papers

May— Inspection Trip.

Birmingham Section

Officers and Committees—1923-24

Executive Committee—W. B. Connally, chairman; F. G. Cutler, vice chairman; A. L. Freret, past chairman; G. H. Fink, J. E. Harrell.

Secretary—W. W. Garrett, Jr.

Papers Committee—L. M. Smith, chairman.

Entertainment Committee—J. E. Harrell, chairman.

Membership and Attendance—M. M. Argo, chairman.

May 17, 1924—

Annual outing.

EDITORIALS

BALL AND ROLLER BEARINGS

THE ball and roller bearing manufacturers are offering to members of the A. I. & S. E. E. through the columns of the Iron and Steel Engineer their reasons for the application of ball and roller bearings to motors in the iron and steel industry. It does not necessarily follow that the reasons offered or the data present are sufficient to provoke any radical change in the application of bearings to motors; however, they offer considerable thought for serious reflection upon the part of the manufacturers of motors and to the users.

Read the articles carefully, also the discussion offered, then form your own conclusions.

ELECTRIC FURNACE COMMITTEE

MR. GEORGE SCHAEFFER, Chairman of the Electric Furnace Committee, solicits your co-operation in the preparation of the Committee's report for the year 1924. The Committee is particularly anxious to secure operating data with respect to the performance of electric heating furnaces, such as heat treating furnaces for tool steels, malleable iron, rivet furnaces, tin pots, galvanizing pots; in fact, any data which any member feels will be of practical value to the iron and steel industry.

Write Mr. George Schaeffer, Electrical Engineer, Carpenter Steel Company, Reading, Pa.

GIANT POWER SURVEY OF PENNSYLVANIA

MR. JUDSON C. DICKERMAN, assistant director of the Giant Power Survey of Pennsylvania, presented a paper before the Philadelphia Section, Friday, May 2, 1924, at Harrisburg. The author presented the tentative program of the Giant Power Survey, which is in part as follows:

(1) Large sized central stations of 500,000-kw. capacity.
(2) These stations to be located at or near the coal mines.
(3) A trunk system of transmission lines, operating at 220,000 volts and possible above that, with large capacity.
(4) An integrated or adequately tied together transmission and distribution system.
(5) The pretreatment before using as power plant fuel of all bituminous coal capable of yielding by-products of value.
(6) Current distributed to the rural population.
(7) Trunk line railroad electrification.
(8) Reduction and equalization of rates consistent with the cheapening of costs by the economies to result from the other planks of program.
(9) The utilization of water powers whenever feasible.
(10) Utilization of inferior portions of coal mined, reduction of waste in mining and possibly the marked reduction of the smoke nuisance through the operation of smokeless fuel production plants in connection with the Giant Power Stations.

The general connections of the steel industry to the Giant Power system as the author conceived it would have the following advantages to the system and to the public interest:

(1) Increase the load factor of the system.
(2) Add somewhat to the diversity factor.
(3) Increase the size of the power units to be added each year to the growing system, thus tending to make them cheaper per unit.
(4) Be a source of power for the Giant Power System in periods of light demand in the steel plants, while yet the blast furnaces and coke ovens were in operation producing surplus gas and other fuel.

On the part of the steel works it would:

(1) Reduce the amount of standby power equipment needed.
(2) Simplify or assist in carrying the short time heavy peak loads.
(3) Permit of operation of all power units at loads of maximum efficiency at all times.
(4) Reduce the amounts of new capital investment necessary to meet the demands of a growing business.
(5) Purchase at lowest possible cost any current needed in excess of that produced from waste heat.
(6) Cheaper power to the scattered quarries and mines producing the stone and coal required.
(7) Removal of some of the density of traffic converging at the steel plants by reducing the carloads of coal now brought in to furnish power.

This paper will appear in the July issue of the Iron and Steel Engineer.

PUBLIC SERVICE COMMISSION DECISIONS RELATIVE TO RATES

IN a paper presented before the Philadelphia Section, May 2nd at Harrisburg, Dr. F. Herbert Snow, Chief of the Bureau of Engineers of the Public Service Commission of the State of Pennsylvania, endeavored in the presentation of his subject to touch upon the salient features of the Commission's decisions affecting electric light and power rates. Out of many scores of such cases Dr. Snow selected ten cases as types, where contracts, rules and regulations demand minimum or service charge valuations, classification and miscellaneous cases featured the Commission's decisions. This resume affords a general understanding of the law and its interpretations by the appellate courts and the practice of the Public Service Commission at the end of its first ten years' existence.

Dr. Snow's paper will be a very valuable one for future reference and no doubt will be placed in the files of all the legal departments of the iron and steel

industry as well as all the engineering departments. The paper will appear in the July issue of the Iron and Steel Engineer.

ELECTRICALLY DRIVEN REVERSING BLOOMING MILLS

OVER 200 engineers, representing many of the leading steel industries in America, attended the "Electrically Driven Blooming Mill" meeting held in Pittsburgh May 10th, 1924, under the auspices of the Pittsburgh Section of the A. I. & S. E. E.

For the first time in the history of the General Electric Company, Westinghouse Electric & Manufacturing Company and Allis Chalmers Manufacturing Company, a joint paper was prepared by these companies which outlined in a most complete and careful manner the design, installation and operation of electric driven reversing blooming mills. It is to be hoped that in the future when a subject as broad as this one is presented before the Association that the same co-operation will be in evidence, as only in this manner will the operating engineer be able to secure a complete analysis.

The first electrically driven reversing blooming mill was purchased by the Algoma Steel Corporation at Sault Ste. Marie, Canada. It was installed under the supervision of Mr. R. S. Shoemaker, who was then electrical engineer of that plant. The second electrically driven reversing mill was purchased by the Steel Company of Canada, Hamilton, Ontario. It was installed under the supervision of Mr. E. S. Jefferies, electrical engineer of that plant. Mr. Jefferies is a past president of the A. I. & S. E. E., and Mr. Shoemaker enjoys the honor of being the president of this organization this year.

Since that time 57 electrically driven reversing blooming mills have been purchased, seven of which have replaced steam engines. It might be well to note at this time that an installation has just been completed where a motor has repaced a steam engine and it required just 18 days for the change. This particular installation was a 40-inch reversing blooming mill. An article covering this installation will appear later in the Iron and Steel Engineer.

Some of the major points brought out by the authors were:

First—The fundamental advantages of electric drives over steam drives.

Second—Definite reasons why the steel companies were purchasing electric driven mills in preference to steam driven mills.

Third—Costs were submitted on a reversing mill driven by a motor which covered a period of 11 years.

Fourth — Graphic charts were shown which gave hourly readings of tonnage, elongation and power consumed.

Fifth—Figures were presented which proved that electrically driven reversing blooming mills are capable of producing tonnages equal to any of those driven by steam engines.

Mr. C. H. Hunt, Chief Engineer of the Weirton Steel Company, at Weirton, W. Va., presented one of the most elaborate and complete discussions ever presented before any engineering society. His discussion covered the operation of a twin tandem compound condensing engine which drives a 40-inch blooming mill. The discussion included the operation of the boiler plant. The complete paper, together with all the discussion, will appear in the August issue of the Iron and Steel Engineer. This will be one of the most complete treatises which has ever been presented covering this subject. The following engineers participated in the discussion:

E. S. Jefferies, Electrical Engineer, Steel Company of Canada, Hamilton, Ont., Canada.
Gordon Fox, Electrical Engineer, Freyn, Brassert & Company, Chicago, Ill.
C. H. Hunt, Chief Engineer, Weirton Steel Company, Weirton, W. Va.
Barton R. Shover, Consulting Engineer, Pittsburgh, Pa.
F. G. Gasche, Consulting Engineer, Donner Steel Company, Inc., Buffalo, N. Y.
R. H. Keil, Power Engineer, Jones & Laughlin Steel Company, Pittsburgh, Pa.
Wm. Fonstrom, Chief Engineer, Wisconsin Steel Company, South Chicago, Ill.
Mr. Ivensen, Chief Engineer, Mesta Machine Company, Pittsburgh, Pa.
James Farrington, Superintendent Electrical Department, Wheeling Steel Corporation, Steubenville, Ohio.
A. W. Duncan, Superintendent Electrical Department, Weirton Steel Company, Weirton, W. Va.
George E. Rose, General Superintendent, Wisconsin Steel Works, South Chicago, Ill.
John H. Bode, Mackintosh, Hemphill & Company, Pittsburgh, Pa.
F. M. Van Deventer, Power Engineer, National Tube Company, Pittsburgh, Pa.
R. H. Stevens, Chemical Engineer, Bethlehem Steel Company, Cambria Works, Johnstown, Pa.
A. L. Foell, Chief Engineer, Donner Steel Company, Buffalo, N. Y.
R. L. Shoemaker, Superintendent Maintenance, American Rolling Mill Company, Middletown, Ohio.
E. Friedlaender, Superintendent Electrical Department, Edgar Thomson Works, Braddock, Pa.
E. O. Knight, Mechanical Engineer, C. S. Co., Clairton, Pa.
F. A. Wiley, Electrical Engineer, Wisconsin Steel Company, South Chicago, Ill.
Frank McGough, Steam Engineer, Weirton Steel Company, Weirton, W. Va.
W. C. Buske, Master Mechanic, McConway & Torley Company, Pittsburgh, Pa.
J. D. Wright, General Electric Company, Schenectady, N. Y.
Rae W. Davis, Allis Chalmers Company, Milwaukee, Wis.
Edwin H. Brown, Engineer, Allis Chalmers Company, Milwaukee, Wis.
G. E. Stotz, General Engineer, Westinghouse Electric & Mfg. Company, East Pittsburgh, Pa.

INSPECTION TRIP

OVER two hundred engineers representing many steel industries in the United States attended the Harrisburg meeting, held under the auspices of the Philadelphia Section, May 2nd and 3rd, 1924. The inspection trip through the Steelton plant of the Bethlehem Steel Company at Steelton, Pa., was under the general supervision of John C. Reed, electrical engineer of this plant, and an active member of this Association.

Beginning at 9 A. M., Power Plant No. 1 was visited. Next the machine shop, then the 44-inch mill, after which the 35-inch, the 28-inch, No. 2 power plant, ingot stripping yard and open hearth departments were visited. Next the open hearth, 14-inch and 16-inch mills, No. 1 forge department, gas blowing engine and Nos. 3 and 4 power departments were visited. Next the blast furnace department, merchant mill, steel foundry, frog and switch department were visited, after which the coke ovens, east end pumping plant, bridge and construction departments were inspected.

Guides were provided for each group of 15 visitors, and every effort was made to explain the policies and processes of this company. One of the outstanding features which impressed the visiting engineers was the efforts and methods used to prevent accidents in this plant. The welfare conditions at this plant were also exemplified by the cleanliness of all of the departments, the well lighted buildings, the heating and ventilating systems in use at the maintenance shops. A complete description of the departments visited is described later in this article. A luncheon was served at 12 o'clock by the Bethlehem Steel Company at their general offices.

At 2 P. M. the visiting engineers boarded street cars and were taken to the plants of the Central Iron & Steel Company and Harrisburg Pipe & Pipe Bending Company at Harrisburg. Guides were provided at these plants for each group of 15, and many interesting features were noted in the steel making processes.

In the evening at 6 o'clock, an informal dinner was held in the ball room of the Penn Harris hotel at Harrisburg. The entertainment for this dinner was furnished by the Bethlehem Steel Company. The music by the employes' band of the Bethlehem Steel Company was one of the most delightful incidents in connection with the day's proceedings. Mr. C. C. Robbins, General Manager of the Steelton plant, is to be congratulated on having at his command an organization capable of taking its place amongst the leading bands in this country.

At 8 P. M. a technical session was held in the House of Representatives, the state capitol of Pennsylvania, at which time Mr. Judson C. Dickerman, Assistant Directory, Giant Power Survey of Pennsylvania, presented a paper covering the activities of the Giant Power Survey of the State of Pennsylvania. Dr. F. Herbert Snow, Chief of the Bureau of Engineering, Public Service Commission, of the Commonwealth of Pennsylvania, presented a paper covering "Decisions of the Public Service Commission of the Commonwealth of Pennsylvania." Both papers were followed by an informal discussion which was very interesting.

On Saturday the visitors were taken to the Metropolitan Edison Company's 200,000-kw. power plant now under construction at Middletown, Pa. The station is designed for an ultimate capacity of 200,000 kw. and to cost approximately $20,000,000. The first unit of 30,000 kw. capacity, costing about $4,000,000, is expected to be in operation some time in October.

The executives of the steel industries at Harrisburg and Steelton, and the officers of the Philadelphia Section are to be congratulated upon the thorough manner in which the program was carried out, and it goes without saying that the engineers who availed themselves of the opportunity to visit these modern steel plants will take back home many ideas which will be of practical value to their individual companies.

Outline of departments in the Steelton plant in order in which they were visited by the Association of Iron and Steel Electrical Engineers:

Entered plant at the Locust Street gate.

No. 1 Power Plant

1—Allis steam blowing engine for blast furnace.
2—1500-kw. engine driven generators, 440 volts.
1—1500-kw. mixed pressure turbine, 440 volts.
3—300-kw. rotary converters, 250 volts, d.c.
9—300-kva. step-up transformers, 440 volts to 11,000 volts.
3—500-kva. step-up transformers, 440 volts to 2200 volts.
3—300-kva. step-up transformers, 440 volts to 2200 volts.

Machine Shop

140 feet wide by 600 feet long. This shop does most of the repair work for the plant and rough turns forgings for outside orders.

The crane equipment, including outside runway, consists of:

2—50-ton cranes.
2—30-ton cranes.
1—20-ton crane.
4—10-ton cranes.

The Electrical Repair Shop is located on the gallery of this building.

44-inch Mill

Gas producer for 44-inch mill furnaces; 1—7½-ton grab bucket crane.
20—Soaking pits with electrically operated covers.
1—20-ton ingot crane.
1—7½-ton ingot crane over soaking furnaces.
Operating stage for 44-inch mill. Two men operate this mill.
No. 3 Sub-Station and 44-inch Mill Motor Room—
1—5800-hp. continuous rating reversing mill motor.
1—Flywheel motor generator set for operating above motor.
3200-hp., 6600-volt Motor, 600-volt d.c. Generators—
Slip regulator for 3200-hp. motor.
3—1000-kva. transformers, 6600 volts to 2200 volts.
3—500-kw. motor generator sets, 2200 volts a.c., 250 volts d.c.
1—80-ton crane over this sub-station and mill.
44-inch Mill Automatic Controller Room.
Shear Building and Shipping Dept. of 44-inch Mill—
1—30-ton crane.

After shearing the steel it is either transferred to the 35-inch mill by rolls or pushed off the rolls and loaded on trucks for shipping.

Electrical Department Office and Storehouse.
Roll Turning Shop.

35-inch Mill

4—Reheating furnaces, one with electrically operated doors.
1—Charging crane.
1—Drawing crane.
2—Bloom carriages automatically stopped at both ends and at rolls.
Operating stage.

Mill is driven by a reversing steam engine; all the auxiliary equipment is motor driven. This mill is operated by two men.

Steel from this mill passes directly to the 28-inch mill.

28-inch Mill (Rail Mill)

Steam driven, 3 high mill; all auxiliary equipment, including the tilting tables, is electrically operated.
1200-hp. wound rotor induction motor on finishing stand used only for girder rails.
Hot saws and hot beds.
Rail finishing department where rails are straightened and drilled.
Crane Equipment of Mill—
1—100-ton roll changing crane.
2—30-ton cranes over mill section.
1—10-ton crane over hot beds.
1—10-ton crane over finishing department.
1—25-ton crane over engines.
Shipping Yard—
2—25-ton cranes, each equipped with two rail handling magnets.
3—Double trolley cranes, each trolley 5-ton capacity.

No. 2 Power Plant

1—2500-kw. mixed pressure turbine, 2200 volts.

This station is connected through transformers with No. 1 and No. 4 Power Stations.

3—Steam driven air compressors and a number of miscellaneous pumps.

Ingot Stripping Yard

1—320-ton Alliance electric hydraulic stripping crane.
1—100-ton Morgan electric stripping crane.

(Continued on page 278)

INDEX to ADVERTISERS

A

Alliance Machine Company 321
Allis Chalmers Mfg. Company 327
Appleton Electric Company 317
Automatic Reclosing Circuit Breaker Company ... 282

B

Baker R & L Company 290
Benjamin Electric Mfg. Co. 291
Bussmann Manufacturing Co. 293

C

Chicago Fuse Manufacturing Company 316
Condit Electrical Manufacturing Company 287
Corliss Carbon Company 314
Crocker Wheeler Company 318
Crouse Hinds Company 311
Cutler Hammer Manufacturing Company 306
Cutter Electric & Manufacturing Co. 330-Cover 3

D

Delta Star Electric Company 301
Duquesne Light Company 299

E

Economy Fuse & Manufacturing Company 289
Electric Controller & Manufacturing Company ... 300
Electrical Engineer & Manufacturing Company .. 325

F

Farrel Foundry & Machine Company 288
Fawcus Machine Company 319

G

General Electric Company .....Cover 4

H

W. F. Hessel Company 324
Ludwig Hommel & Company 284
Hyatt Roller Bearing Company 296

I

Ideal Commutator Dresser Company 316
Iron City Electric Company 324

J

W. J. Jeandron 294

K

Keystone Lubricating Company 281

L

George T. Ladd Company 303
Lapp Insulator Company 315
Le Carbone Brush Company 294

M

Martindale Electric Company 319
Moloney Electric Company 314
Morgan Construction Company 318
Morgan Engineering Company 315
Morganite Brush Company 317

N

National Carbon Company 283
Nichols Lintern Company 322
Norma Company of America 313
R. D. Nuttall Company 305

O

Ohio Electric & Controller Company 286

P

Packard Electric Company 332
Pawling & Harnischfeger Company 285
Pittsburgh Transformer Company 321
Pyle National Company 331

R

Reliance Electric & Engineering Company 329
Roller Smith Company 312
Rollway Bearing Company 302
Rowan Controller Company 325
Rumsey Electric Company 320

S

Shepard Electric Crane & Hoist Company 297
SKF Industries, Inc. 304
Smith & Serrell 308
Square D Company 309
Strom Ball Bearing Manufacturing Company 310
Sundh Electric Company 323

T

Thomas Flexible Coupling Company 307
Thompson Electric Company 326
Tool Steel Gear & Pinion Company 322
Trumbull Electric Manufacturing Company 292

U

United States Graphite Company 320

V

V. V. Fittings Company 295

W

Wayman Electric & Manufacturing Company ... 323
Western Electric Company 298
Westinghouse Electric & Manufacturing Company
.....328-Cover 2

INTRODUCTION
Ball and Roller Bearings

By D. M. PETTY, Chairman
Special Bearings Committee, A. I. & S. E. E.

The subject of applying ball and roller bearings to electric motor has been before the Association of Iron and Steel Electrical Engineers for several years. A number of the steel plant engineers have tried both ball and roller bearing motors and found them very satisfactory. A number of the motor builders are putting motors on the market equipped with ball and roller bearings.

In order that all members of the Association could inform themselves on this subject, it was considered advisable to discuss it at section meetings in Philadelphia, Chicago, Cleveland and Pittsburgh under the title, "Mounting and Maintenance of Roller and Ball Bearings in Electric Motors." A ball bearing engineer and a roller bearing engineer prepared a paper for each meeting and we are indebted to each of them for their excellent work. Judging from the discussion which took place at each of these meetings, the users of motors are rather anxious to find a bearing that will give less trouble directly and indirectly than present bearings, and sooner or later these men will insist on having their motors equipped with ball or roller bearings.

We are familiar with the obstacles which apparently stand in the way of standardizing the mounting dimensions of the mill type motors and the fact that though only fractions of an inch in many cases prevent interchanging these various makes of motors of equivalent size, they cannot be changed to a standardized dimension sheet without a large cost.

To prevent a repetition of this economic crime in the matter of ball and roller bearings, the Bearing Committee of the Association of Iron and Steel Electrical Engineers have been working with a committee from the Power Club and a group of ball and roller bearing engineers, endeavoring to select a group of ball bearings and roller bearings that can be set up as the standard bearings to be used in electric motors.

Such a list was worked out tentatively during the convention at Buffalo in September, 1923. The various motor designers took this list back to their offices and made a study of it. In January, the Power Club Bearing Committee called a meeting with the bearing engineers to discuss the subject further. Since that time the ball bearing engineers have been giving the subject further study. Early this summer, at least a tentative list of standard bearings should be available. When this list is set up the users of motors can then proceed with their program of changing over motors from sleeve to roller or ball bearings; the builders can also lay out lines of motors equipped with ball or roller bearings.

A great deal of discussion has taken place in reference to the size of ball or roller bearings that should be used with a motor of given rating in horsepower and speed. While this is necessary in making a study of the subject, it is not essential to standardization.

The great saving due to standardization will be accomplished if we select a list of sizes; the designers can use any one of these sizes he cares to in his motor; the user will buy those motors which are best equipped, taking into consideration both reliability and price.

As has been pointed out in the various papers, ball or roller bearings must be properly mounted and properly maintained if they are to be of real value in helping the iron and steel indutry produce more at lower cost.

It is the hope of the Bearing Committee that these papers will be of some assistance to those members of the Association who are considering ball or roller bearings for their motors.

Application of Taper Roller Bearings*

By T. V. BUCKWALTER†

I am not very well acquainted with your Iron and Steel Electrical Association but if it is at all comparable with the local section of the Society of Automotive Engineers, it is one of the best sections, and from what I see this evening thus far I am inclined to believe that your section can be considered in the same light.

We had a rather small amount of time to prepare anything for this meeting. As a matter of fact, I got home Saturday noon and immediately took steps to get up here today. I can see that I am going to have difficulty going through with it because I picked up quite a cold last week, and talking is not the best thing for a cold. I will endeavor to show you some of the features in connection with the application of a taper roller bearing and will have to show you the Timken type roller bearing applied to the electric motor.

This drawing is intended to illustrate the application of the taper roller bearing to the electric motor frame. We have taken one that is rated at about 100 to 150 hp., depending on speed.

In the Timken bearing there are two component parts of which the principal part is the cone, so-called on account of its shape and on which is assembled the rollers with the cage for guiding the rollers in their proper position. The second unit is the cup. The two units together form the complete bearing and form what are generally called the two unit type of bearing as distinguished from the radial type of ball bearing which is of a single unit type. Some taper roller bearings are made with the cage and roller assembly independent which forms a three unit type of bearing.

Now the principal feature in a taper roller bearing, is as the name implies, its taper construction so that a bearing of the type outlined here is suitable for carrying a certain radial load, and in addition the two bearings mounted in pairs will take a thrust load and between them any combination of thrust and radial loads. It provides about as simple a construction for carrying any combination of reactions that might be imposed on a bearing as we can obtain.

The lubrication on electric motors is a matter in which there has been considerable experiment. We have had a number of motors running with oil lubrication in which we provide an oil level just below the level of the armature shaft, and that takes care of applications that are run at fairly high speed. A medium grease is quite suitable and is generally used on applications of medium and low speeds.

The most difficult part of a bearing application for electric motors is not so much the question of type of lubricant to use but the matter of retaining the lubricant in the housing. We took up the experiments on the mounting of Timken bearings on electric motors some five or six years ago and have rebuilt quite a number of electric motors with the Timken bearing mounted thereon. It was quite plain from our early experiments that the enclosure was one of the weak elements and we came to the conclusion at an early stage in our experiments that the ordinary type of enclosure which is perfectly satisfactory on automotive work would not be satisfactory on the electric motor work. These experiments as we proceeded further along with them indicated that the spiral groove cut on the cap but not carried through into the bearing housing provided the most certain means of retaining the oil or grease or paste oil lubricant. That of course is satisfactory for operation in one direction and to provide for operation in either direction a similar groove outside of an annual chamber will take care of the enclosure problem with the motor for operation in either direction.

Our earlier experiments with the ordinary felt type of enclosure such as is in general use on automobile work would run from two to six months without any further attention and without a waste of the lubricant, but this was not considered satisfactory and we thought that a motor should run at least six months to a year without any attention from that standpoint.

A motor built somewhat on this order but not exactly in that we did not make the armature shafts new but used the shaft that we found on the motor, has now been running in the Timken plant for a period of about 17 months. Our aim was to run at least 12 months on a duty of rather severe type, in service 22 hours every day, and to continue this test for a year without touching the motor in any way. Just to be on the safe side we carried it a month further to get a lucky number, and ran it for 13 months without touching it.

In taking it down we found the enclosure had leaked out through this joint between the bearing housing and the frame at this point, and one bearing was running practically dry, but the other one was in very good condition. The indications were that if we had taken the pains of shellacing these joints in making the original assembly that bearing would probably have run until the oil was churned up or had disappeared entirely, if such a thing could be. However, that test could be considered as having been entirely satisfactory for over 12 months without any attention to the motor whatever. But, I am anticipating my story a little on the mounting of the bearing.

To get back to the taper construction and again calling your attention to the fact that the bearings are mounted in pairs—the final assembly can be made in the case represented here by detachable cap with thin shims providing for a close adjustment of around three to five thousandths, these shims being interposed between the cap and the motor head. The shims were introduced at this point to facilitate taking up adjustment for wear at any future time. This adjustment can be made by removing the cap, removing one shim of the thinnest generally, and replacing the cap which will draw the assembly together possibly three to five thousandths of an inch which would be the normal amount of wear and which might or might not require adjustment, depending on how the motor is being maintained.

*Presented at Cleveland Section, January 14, 1924.

†Chief Engineer, Timken Roller Bearing Company, Canton, Ohio.

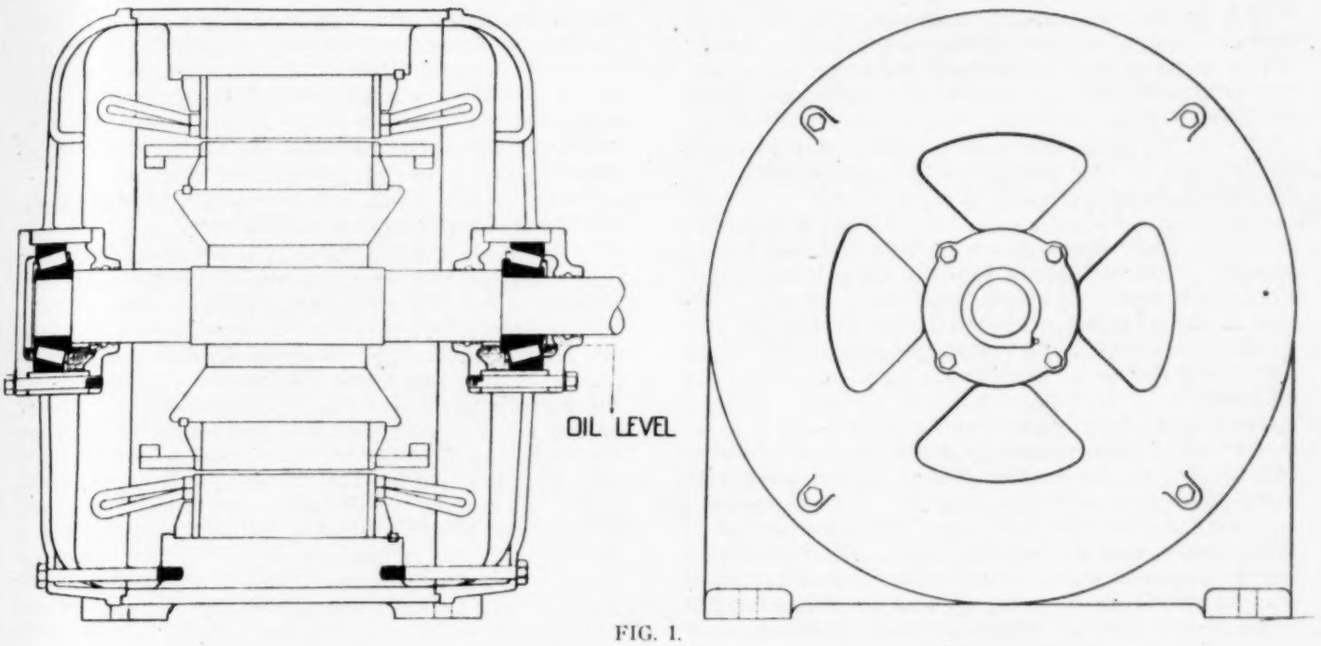

FIG. 1.

Now you will find a good many differences of opinion, and we may as well mention it at this time, as to the need for an adjustment in a bearing, and we might state on that subject that the bearing does not wear due to any faults in the bearing but the wear is due to the slight amount of dust and grit that gets into any enclosure and which with the millions or hundreds of millions of revolutions that a motor runs, which would involve a good many miles of rolling friction, will gradually wear away infinitesimal small amounts of the bearing metal.

It is an unfortunate fact that ordinary dust consists largely of silica. Silica is practically sand and is an oxide of silicon, and it so happens that silica is harder than hardened steel, and a bearing running in the presence of even a slight amount of silica, such as might be on any object that you pick up, will gradually wear away. And we consider it is advantageous to provide a simple ready means for taking up that wear.

Another point that naturally follows from the adjustable feature is that any desired degree of looseness can be obtained on the assembly. For instance, it might be found (and I am speaking largely now from automotive experience which I think can be applied to electric motors) that a running clearance of possibly a half-thousandth might produce the point of maximum freedom from vibration or minimum amount of noise or it might so happen that the various weights and masses are such that a slightly greater amount of running freedom or slightly less is desirable. This is readily determined by experiment and in assembly it is almost instantly obtained by a competent, skilled mechanic.

The question that naturally arises is, "How long will these bearings last?" That we can't say because in the automotive industry where we have most of our hundred million bearings running, the bearing generally outlasts the car. There are a number of instances where cars have run 200,000 miles with the original bearings. We operated the bearings on these electric motors somewhere in excess of 500,000,000 revolutions and at that service we have had wear end-wise on one of about five-thousandths in drop of the bearing which would mean about one-thousandth in diameter of the bearing; in another test of somewhere around 400,000,000 revolutions we had a drop of one-thousandth. One bearing which we operated for about six months continuously at a speed of 2400 and 24 hours a day, we had a very slight amount of wear, hardly measurable. Those, of course, are conditions in which we have taken care to obtain a clean assembly free as possible of dirt and grit.

In passing I might state that all the dirt and grit that gets in the bearing is not going to be in the original assembly. They may be introuced through infiltration of the air through the enclosures or through any leakage point, or mostly through the oil. Our friends engaged in the manufacture of lubricant, I don't think, have ever succeeded in getting an oil that is entirely free of foreign matter and unfortunately much of the foreign matter is sand or grit.

The point that could be mentioned in connection with an anti-friction bearing on electric motors is the economy of space lengthwise. There is very little projection over the head of the motor with consequent reduction in spacing between the bearings, resulting in more rigid support of the rotor.

I think you have all had that matter brought to your attention. At least, I had it brought to mine when I was in your position on the buying side some 10 to 20 years ago, and I believe it is generally recognized as an inherent advantage of the anti-friction bearing.

Coming now to the end view the mounting of the bearing is practically symmetrical and I believe provides maximum possible ventilating space in the motor head, and should, at least I am so informed by some bearing experts, provide the best running conditions as regards cooling of the motor, particularly under heavy duty. This application of the bearing is made to scale on an outline which I believe is in current production by one of the larger companies, and the gain in air space is quite material.

The second advantage following from the symmetrical design of the bearing is that the assembly can be made in the factory and motor set up in any way either side-ways or vertical without removing the end heads.

I want to point out now a feature not so much connected with the bearing in its application to the motors but one we consider as a rather important consideration in bearing selection and that is in regard to the selection of bores—the bore of the cone for the bearing. We understand there is quite a movement on foot to adopt a standard, based on the original German sizes originated with the D. W. F. Company in Berlin possibly about 18 or 20 years ago, and in which the increment between adjacent sizes varies by five millimeters. It so happens that this variation between sizes (practically three-sixteenths of an inch) is satisfactory when they get up to bores of about three inches and up, but in the smaller motors commencing with three-quarters to one inch cone bore and shaft diameter we consider that this variation is entirely out of the question and abnormally high. We have found that to be the case in the automotive industry and we would say in regard to that general condition that the taper roller bearing manufacturers, especially ourselves, believe that the industry requires a series of bearings varying in bore by about one-sixteenth of an inch, and we will be prepared to supply them on that basis.

I might state that the strength of the shaft in adjacent sizes, based on the German system from one inch to one and three-sixtenths is about 95 per cent between jumps. We consider that is too high in consideration of the demand for economic production at the present day. These steps gradually are reduced from about 95 per cent to about 20 per cent at three inches, commencing at the 95 per cent at one inch so that we get up to the three inch bore before we get into reasonable increment between adjoining sides.

We might as well grant at the start that an anti-friction bearing is going to cost more than a plain bearing and you gentlemen in using the bearings will not be justified in using them unless you have some consequent economies resulting from their use.

We believe one of the first is the economy in lubrication. As stated already, we have had bearings running a year without attention on the score of lubrication, and you can appreciate, considering the small size of the housing, that the amount of lubricant in a motor of this kind is a rather small item.

In addition to the cost of the lubricant is the absence of attention, which will be a very important factor.

Probably one of the things that will be asked this evening will be the relative friction between taper roller bearings and plain bearings. That is a rather, I am going to say frankly, difficult question (anticipating your question if we can) to answer. I will say first that a very important advantage is that the starting friction on an anti-friction bearing of either the roller or ball type is very little different from the running friction, so the starting friction on your motor will be very greatly reduced. The running friction will depend largely on the character of the lubricant and the accuracy with which the bearings are made and the care with which they are mounted. Assuming that a motor is loaded to about full capacity the energy absorbed in the bearings should not be more than about fifteen-hundredths of 1 per cent, in other words, the bearing efficiency on a fully loaded motor should be around 99.85. I think some of the bearings of other types might run a little higher than that and some of these may run a little lower, depending largely on the type of mounting and the character of the motor.

In any event, it will be appreciated that the running friction is reduced to a very small item.

The rotor on motor illustrated weights somewhere in the neighborhood of 500 pounds. I might state that we have conducted a number of tests or readings on a deceleration wheel that was designed to weigh about one-quarter the weight of an automobile. It so happened to work out about 840 pounds. Our tests on the deceleration wheel which happened to be mounted on two bearings quite similar in application to what you see here, was to speed up that wheel so that the rim velocity was about 60 miles an hour, and let it coast to zero. A bearing under good operating conditions would coast about 60 minutes which indicates that if we could, as far as bearing friction is concerned, accelerate an automobile to a speed of about 60 miles an hour and neglecting all friction except bearing friction it would coast 30 miles, or almost from Cleveland to Akron.

We have mounted a variety of plain bearings on this same decelation wheel and some of them have run three minutes, others six minutes. I believe that the length of run on this test largely depends on the condition of the lubricant. A shaft just moist with lubricant was not very efficient. My own belief is, I haven't had it proved, that a plain bearing running under oil pressure, such as we are getting on automobile crankshafts, so the shaft floats on an oil cushion under pressure of about 30 to 60 pounds, would approach the best taper roller or ball bearing in absence of friction. Of course, it isn't possible to maintain those ideal operating condition on the plain bearing. The unfortunate feature is that the failure of the film of oil will increase the friction very materially, probably a thousand per cent.

It takes a long time to work out some of these details on bearing friction, and as a matter of fact we don't get on it except as we are compelled to because we have so many other problems more pressing.

This drawing shows a rather wide air gap and while I am not an expert on motor design I am under the impression and have been told by those who I believe to be expert that maintenance of uniform air gap is one of the highest requisites in electric motor operation. I think it will be granted that the anti-friction bearing is going to maintain a much more nearly uniform air gap than the plain bearing. This is largely true for the reason that the wear on the cup is practically negligible and the wear on the cone and rollers is uniformly distributed so that the drop in the armature shaft is a very small item. Our tests thus far have indicated that after 12 months of constant operation under full load on a motor approximately this size that the drop in diameter is only about one-thousandth or the difference in the air gap would only be about a half-thousandth measured on the radius. It is believed that here is a method of maintaining air gap uniform and if any advantage is to be gained or obtained in running under thin air gap it would be obtained to the maximum extent in the anti-friction bearing.

These tests that have been carried on and on which I have reported here haphazardly, have been on General Electric motors of about 60 hp., I think 1200 r.p.m., and Westinghouse of the same power at 900 r.p.m., together with a small Fairbanks-Morse motor and a motor built by a small independent company. They all produced very satisfactory results as regards bearing operation.

I might say in passing in regard to these various tests that we have made on electric motors that a very nicely finished roller bearing is necessary to give you the satisfactory service and long life that we think will be demanded in the electric motor field. The limits are not beyond the realm of commercial practice and involve an accuracy in the roll of about three-tenths of a thousandth as regards out of roundness and a slightly greater figure, about five-tenths, on accuracy of taper. The corresponding accuracy of the cone must be closely approximated. The run-out on the cup is not so important a matter because it means a slight variation only in the air gap. However, our experiments indicate that much care is necessary on the roller and we think that facilities are available to give the bearings for electric motors that necessary care. The service on electric motors will compare very closely as regards standards on noise and accuracy with that on transmissions of automobiles or the pinion or worm shaft on automobile axles or motor truck rear axles. These are the points that are run at motor speed and on which the greatest complaint of noise comes and where irregularity in the bearing is most quickly apparent.

No doubt you would be interested in a comparison between the costs of the roller bearing and your plain bearing. Unfortunately, I am not in position to give you costs on plain bearings but I can make this statement, that you can't measure your cost by direct manufacturing or original costs. I am speaking on very solid footing in this respect, because for many years I kept up the motors and electric machinery in a large railroad shop, in fact, one of the largest in the world, located at Altoona, and when I talk about motor troubles I know pretty nearly what I am talking about because I have been through it, and I say in view of that experience and that situation that the first cost of a plain bearing is not the only cost because you must consider in the life of the motor the frequent renewals and consequent heavy expense, particularly labor expense, and loss of use of the motor in renewing plain bearings. I might give you a few examples on that score.

One of our problems was that we had to maintain at Altoona a foundry in which there were about 40 cranes, as I remember, and the cost of keeping up the motors on them was terrific, but we carefully analyzed the situation and found that nearly all of our costs was due to rubbing of pole pieces, burning out of windings. This was all traced back to wear of the plain bearings with consequent dropping of the armature. If I remember my figures right, the cost of maintaining the motors in that installation ran about 40 per cent annually of the original cost of the motors.

I have always believed that somewhere around 90 per cent of that could be eliminated on a bearing that stayed put. I can't tell you just what the plain bearing costs are. No doubt you know. I might state that the cost of the taper roller bearings for the motor outlined here which might be called 100 hp. at 1200 r.p.m. would be somewhere in the neighborhood of $12.

You can see that it is a commercial proposition to use such a bearing and I believe will be direct economy in bearing operation. I don't claim that there will be direct economy in the original manufacturing cost of the motor because possibly the plain bearings cost less than that, but I think in the long run counting the years the motor should run, the bearing cost will be much less on the anti-friction bearing of proper design and mounting.

I want to say a few words about the facilities of the company with which I am connected for producing bearings. This company, as Mr. Roemer no doubt can substantiate, is located at Canton and I believe is the most complete manufacturing organization in this country. I don't know of any that is more complete.

It involves a complete manufacturing organization from the raw materials, from the ores, so to speak, consisting of a steel mill, rolling mills, tube mills, wire and rod mills, so as to manufacture the component parts for the roller, cone and cup with just enough machining to provide necessary clean-up; and all of the various cups and cones are made from tube stock. The tube is made with just enough finish to clean up so the company is not dependent on any outside source of supply of any kind except the raw material. The raw material is probably one of the simplest things to provide. You might say the bearing is made from the ore but that isn't strictly true because it is made from scrap, scrap material of certain qualities — any quantity of scrap of just the right kind to make a high class anti-friction bearing steel is readily available. It is melted in an electric furnace with the refining processes available in that furnace with regard to keeping down impurities and controlling the carbon and alloy content within very narrow limits, and it produces a bearing that has the utmost in resistance to wear and in life resistance to fatigue and general good bearing steel qualities.

In addition to these facilities for producing raw material this company has developed almost all of its machinery for manufacturing the complete bearing. These include the semi-automatic machinery for finishing roller, cones and cups, automatic machinery for hardening, automatic grinding machinery. About the only thing we haven't been able to get automatic or semi-automatic is the assembly. We still depend on the human element for various assembly and inspection operations.

The main plant is located at Canton. It has a capacity of about 55,000 bearings a day. In order to provide a second reliable source of supply a second slightly smaller plant located at Columbus, Ohio, which has about the same general equipment as the parent plant except that it does not have the facilities for producing the raw material. This is all done at Canton.

You gentlemen are interested in electric motors primarily and I might say that the Canton plant is almost entirely driven electrically. The billets are rolled in a 22 inch electrically driven bar mill. The tube is made in one of the largest piercing mills in the country which happens to be designed in Canton and built in Ohio. The rod and wire mills consist of a roughing, intermediate and finishing mill and they are all electrically driven.

While the quality of steel for any purpose particularly in a bearing is largely controlled by the conditions under which it is melted, rolled and finished,

I would say on account of the advantages of the electric drive in these various mills the speed of the mills can be varied to suit the conditions under which the steel is being rolled or fabricated so that a uniform working temperature that varies about 150 deg. is maintained, and niggerheads and hardspots and some of the spots just due to sloppy methods in steel making, are almost entirely eliminated.

That is the one way that we feel that we can be sure of a uniform product in our raw materials, that is, to make it practically all ourselves and control the temperature in all of the various operations within very narrow limits. If you control your rolling temperatures within limits of 150 deg. your steel will not be absolutely uniform but sufficiently uniform for commercial purposes with a view to subsequent machinery operations. That is one of the chief reasons that we had to go into the manufacture of the steel ourselves, because we could not obtain the reliable source of supply in the large variety of sizes necessary from any outside source.

In closing I want to say a word on the general economic aspects of the present industrial situation. This has a bearing on electric motors as well as antifriction bearings and the automotive industry. This country of 110,000,000 people is, as you know, one of the most important, in fact, the most important country in the world and most important industrially. When we say "industrially" we must consider that large percentage of these people, roughly about 35 to 40 per cent are engaged in agriculture. Within the last three to four years this very large percentage of around 45,000,000 people have had a very material cut in their wages or incomes due to the lesser returns they get for their product now as compared with what they received during the war and as compared with what they got before the war when measured against the cost of other commodities.

Now for several years we have believed, and are still working along that idea that the industrial element in this country (and by that I mean the transportation, steel making, automotive, motors, electric and otherwise, and all the other various industrial enterprises) must adjust their product to this large market which is practically 40 per cent of our population, and that means that we must get the cost of our product down somewhere within reason so that the large agricultural element can purchase as freely as before the war

There are very few industries that have done this. The tire industry is now selling tires for about one-third less per mile than the cost of tires before the war. The automobile industry, with which you are familiar, is selling their product at a lower cost today than prewar. We believe it is necessary for every important industry to do that but I don't believe very many of them have seen the light in that respect and they are not making the efforts to get their product down in cost so as to get their portion of the $40,000,000,000 or $50,000,000,000 a year that is being spent in this country. The automobile industry has done that and they are getting their portion which amounts to about $7,000,000,000 a year of the total expenditure of the American people. And a good many of the other industries say that the American people are wasting their money and spending it on automotive products. They are spending it where they get the greatest value for their dollar.

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*Application of Anti-Friction Bearings**

By F. W. GURNEY†

AS I have not had time to prepare a paper I will have to give you what I have to offer in the form of a few remarks. A week ago Saturday I presume quite a few of you gentlemen were present at the meeting in Philadelphia when ball bearings and roller bearings were discussed by Mr. Crow from the New Departure Company, and by a representative of the Hyatt Company. Mr. Crow gave a very interesting discussion on some of the fundamental principles of ball bearings. I am sorry he is not here tonight. I will just make a few brief remarks on the fundamental reasons which control the elimination of friction and incidentally heat and wear from rolling bearings. These will apply to Mr. Buckwalter's type of bearings as well as ours.

I am sorry I haven't any slides, but we will call this the outer ring, what Mr. Buckwalter calls the cup, and this is the inner ring, what he calls the cone of a roller bearing. This is an elemental bearing, and I will say that the ideal elemental frictionless bearing would be a roller bearing, one in which the rolling elements were not spheres or ball, but cylinders. If we could live and do business in an ideal condition, why, we could use such a bearing, but inasmuch as we have to work in by no means ideal conditions such a bearing cannot be used.

If all of these elements were cylinders, a cylindrical ring here, a cylindrical ring here, cylindrical rollers rolling between these two rings, all of them straight, perfectly so, and of perfect fit, all of them incompressible, none of which conditions can ever exist, we would have a bearing which would run practically without friction, without wear, would run almost indefinitely; but inasmuch as these conditions can only be approximated, none of these things are perfect straight cylinders, the rolls will get a little bit out of alignment, instead of staying in these cylindrical elements they will roll out.

Mr. Buckwalter's company has solved that difficulty by making these rolls conical—maintaining the location of the rolls upon the inner ring by ridges or shoulders or beads upon its edge, and holding the outer ring in place in its assembly. With ball bearings we have made the rolls spheres and solved the difficulty by various methods of maintaining the assembly by running these spheres in grooves, an inner in the outer ring and an outer groove on the inner ring.

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†Chief Engineer, Gurney Ball Bearing Company, Jamestown, Ohio.

The balls are made of a slightly shorter radius than that of the curvature of the race grooves. If the steel of the balls and rings were incompressible, which is never the case, and the curvature of these elements was perfect, the contact of the roller elements in the raceways would be mere points. Here again is an ideal condition that is never realized. The steel, although it is the hardest material out of which we can make such things, is still a very elastic and compressible substance, so that the so-called point of contact does not exist. We have instead a spot or an area of contact. In the inner ring the spot of contact due to the compression of the steel under load is quite a considerable arc. In the outer ring the spot of contact is a somewhat lesser arc. In the inner ring the spot of contact is a very elongated elipse. In the outer ring the spot of contact is a less elongated elipse. But we try to make the relative curvature of the two raceways so that we get about the same area of contact on the outer and the inner raceways so that the stress of metal is about the same in either case. The slipping action due to the fact that the ball is rolling on a zone whose circumference is less on the edges than in the middle is one of the sources of friction in the ball bearing. This, however, is not so great a source of friction and so great a source of heat and so great a source of wear as might be imagined. All of those factors are practically negligible.

I think that undoubtedly the principal cause of the heat generated in a bearing either of the ball or roller type is due to what might be called the molecular friction of the steel. If you will consider any point of contact on a ball and trace the action of that point as it goes around you will have what we call a hypocycloidal curve against the outer ring. That is, this point on the ball as it rolls, swings from point to point, merely touches the steel at this point and then immediately leaves it, touches again, leaves and touches, etc. It does not slip when it touches, so that in the rolling bearing we have motion, free motion, with stationary contact. We have motion without movement—a seeming paradox. The contact between the rolling elements at the instance in which they touch each other is stationary. Theoretically the condition of the roller's contact is the same when it is standing still as when it is going. That is the reason there is so little friction and so little wear, so little heat and such long life.

But the steel, to exaggerate it—when the ball touches this raceway, it dents into the raceway, and the raceway dents into the ball. The line of contact, of course, is about the mean between the two deformation lines. And this also happens, there is a little wave piled up in front and behind the rolling element. The steel at this point is going up and down as the ball rolls over, and that causes friction, just as when you take a bar of steel and bend it, it gets hot. The deformation of the steel is the principal action which causes the heat in a rolling bearing.

Of course, there is also the friction between the rolling elements and the separating elements, but that is comparatively slight if the bearings are properly lubricated. Then, of course, there is the friction, as I mentioned before, due to the difference in circumferences of the different parts of the rolling element, but that is so slight as to be practically negligible. The great thing that causes the heat and causes the wear and causes the breakdown is the continual distortion that is going on all the time, and that it is which limits the load carrying capacity of the rolling bearing.

The pressures which we get under the balls on these little areas between the balls and their raceways are sometimes perhaps as great as 500,000 pounds to the inch, way beyond what we are accustomed to regard as the elastic limit of the steel, and yet the steel stands it. How can we account for that fact? I think it is perhaps in this way: The steel under this very intense pressure, under the spot of contact may be considered as a fluid. It is entrapped by the contact at the edges of the spot of contact, so that the steel cannot flow out because it is held back by all of the metal around to contact area.

Here is another interesting fact: The available wearing and stressed surface of the balls is from 2½ to 4 times, depending upon the number of balls in the bearing, greater than the used area of the raceways. In spite of that seemingly very great advantage which the ball has in wearing surface, laboring surface, over the raceway, it is almost invariably the rule that the ball breaks down before the raceway does, both made out of equally good steel, the ball, however, having another advantage in that it is really much better finished, of finer and more accurate surface, than that of the raceway. And still the finer, better element, having a larger working surface, stands the punishment not so long as the poorer surface of the raceway. Why is it? It is because the ball is working on a very convex surface, it is the point of a tool, so to speak, and the raceway is a concave surface. It has better support by this wall around the edge of the little puddle of fluid metal, if you may consider it such, on which it is working. The surrounding metal on the raceway is better disposed to hold back the stressed metal within it than is the case with the ball.

Another important inference from these facts is that the factors of safety which we have to consider very seriously in the design and application of ball bearings cannot be applied in ball bearing as you apply them in other engineering. Mr. Buckwalter, and every engineer that is manufacturing bearings of any kind, knows all too well that there is no such thing in existence, in spite of what he said about his own steel, as perfect steel. The steel in all bearings is full of imperfections.

Now a slight flaw, a little occluded slag, a little occluded gas, a little seam, doesn't amount to very much in a beam, or in a shaft. In the shaft such a thing is usually drawn out in a long streak or seam. In a roller, however, we get this intensified stress right on that imperfection, which is usually invisible, and that causes serious mischief. It has to be a very considerable flaw or defect or seam in the beam or in the shaft to cause any uneasiness, so that a factor of safety of say four in bridge construction or anything like that, is all right. But no such factor of safety will do in a bearing, though the seeming factor of safety, when I speak of 500,000 pounds to the inch, is much less than that. The actual factor of safety, when the steel is buttressed by this wall around the spot of contact, is much greater. At this point I want to protest against the tendency among manufacturers, which I think now is happily passing away, of trying—different competing manufacturers—to beat each other to it in recommending smaller bearings and bearings of less cost for the sake of getting the business. It is a very short-sighted policy.

I am sorry I couldn't have had some slides or some pictures that I could show you, but I have brought along here some of the illustrations which we have gotten out. We show here in these illustrations our methods of mounting motors. You gentlemen are probably very well aware of the fact that the general adoption of ball bearings by motor manufacturers is perhaps being forced upon the trade somewhat unwillingly. We cannot belittle the fact that in the years that are past, in the 20 years that have elapsed since they first began to use ball bearings, there has been an immense amount of trouble in the use of ball bearings in motors as well as in other fields, due to the ignorance that had not been dispelled by experience.

These recommendations here are the result of perhaps 20 years of experience in the application of bearings to motors and other equipment. In here we also give a rather full schedule of the sizes which we recommend for the application of these bearings to motors, and although these sizes are somewhat larger than are often recommended, I will say for the company which bears my name that it is standing pat on these sizes. And if any one cannot see his way clear to use sizes as large as these, we are perfectly willing for him to gain a little experience by using smaller sizes at the expense of some other manufacturer.

I think this meeting can be made the most useful to all of us if we can have a general exchange of opinions. If you would make an easy mark of me for questions, to be fired at me, I will try to answer them. If I can't I will be very free to say I can't. There is a great deal that has transpired in the last 20 years in the bearing business that I haven't learned.

The first illustration shows a horizontal motor mounted on a double row ball bearing with a housing so that the entire mounting can be taken off without letting dirt be introduced into the bearing. Right here I want to very heartily second what Mr. Buckwalter said about the introduction of dirt into bearings. I will even go stronger than he went. The wear of a ball bearing, and I guess a roller bearing also, is almost invariably due to dirt of one kind or another in that bearing. In 20 years' experience we have developed a great deal in the way of means for excluding dirt from bearings and retaining the lubricant. If the dirt can be entirely and effectively kept out of the bearing the wear is almost negligible, almost imperceptible.

I have in mind a case. A number of years ago we put a ball bearing in a testing machine where we could keep out the dirt. It was a test we were making for an axle manufacturer for a three-quarter floating rear axle. On this bearing, which was to carry one-fourth of the weight of the car normally, we put the full weight of the car, the whole car on one bearing. We ran this bearing at a speed of 3½ miles a minute for a distance of something over 100,000 miles, and that bearing at the end of that test showed no perceptible, no discernible wear. It was still just as tight as when it started out on that run.

Why is it that you never get such a case in actual service? For the simple reason that you don't keep out the dirt. That is all.

There is also another source of dirt than those mentioned by Mr. Buckwalter. Not only does it come from the outside, but dirt is made in the bearing itself. This shows the importance of proper mounting. If there is any movement between the inner ring or the cone and the shaft, there is an abrasion between the steel of the ring and the steel of the shaft, making an impalpable powder of steel or iron which almost immediately, in spite of the oil of the bearing, becomes oxidized, and you know what you have then. You have oxide of iron, or rouge—a beautiful abrasive, but a darn poor lubricant. And this abrasive soon charges the lubricant. You have seen bearings where the lubricant became red. That means simply oxide of iron. Your lubricant is full of rouge, and the wear becomes destructive and very rapid when that occurs. That is very common. The same thing occurs in much worse shape when you get the wear from gears or anything like that into a bearing, which is very frequently the case.

But if the bearing is mounted, if the inner ring is put upon a shaft, we will say, with a press fit, and held further by a nut—the more severe the service the more important is the tightness of the press fit and the clamping—why, the wear will be controlled, the generation of rouge, of oxide of iron in the bearing will be prevented, and you will have an indefinitely longer life to your bearing than will occur if your bearing is loosely and movably mounted on its seat.

We have a very successful method of lubricating the bearing with grease. On the cap which encloses the bearing on the pulley, or gear, or pinion side, we have a grease cup at the top and a drainage cup at the bottom. You can fill this cup at the top and the grease can only get into the drainage cup at the bottom by going through the bearing. If the proper grease is used that will go very slowly.

I will say that in our own plant at Jamestown we are running with an electric drive throughout the plant, using Niagara power, and we are using principally oil in our motors there. We have motors there that have been running for seven years. During the war, about four or five years of that time, we were running night and day. Those motors were lubricated once a year. Sometimes they were inspected twice a year. We simply don't know that there are any motors there. There has not been a case, I think, of a bearing wearing out during that time. We have had one or two cases where one of the retaining nuts came loose and caused a little trouble, but for actual wear, it has not yet occurred so as to be objectionable.

Another very important feature, advantage, of the rolling bearing over the sliding bearing is that in case the lubricant fails, either through negligence or other cause, disaster does not immediately occur. I remember we had a case in the Pierce-Arrow plant where there are a great number of polishing machines that were mounted by Mr. Fred Kent on ball bearings. They were run at a speed of 3,000 to 4,000 rpm. on No. 310 bearings. They are lubricated with grease. Mr. Kent gave instructions when those machines were set up to fill all the housings with grease, and every six months they were to be opened up, the grease cleaned out and new grease put in. At one time when they inspected a bunch of machines that had been set up six months previously they found one machine perfectly dry. They had neglected or failed to put the grease in the housing. That machine had been running six months, night and day, at a speed of 3,000 to 4,000 rpm. without a particle of lubricant in it, and nobody knew it. The bearing was somewhat worn, but it was not worn

enough so that anybody knew that anything was the matter with it. I don't think they took it out. They filled it with grease and it may be running yet for all I know.

If that had been a plain bearing, instead of running six months dry it would not have run more than six seconds or at most six minutes, and the machine would have been out of commission. Of course, we don't recommend the running of ball bearings without lubricant, but still it is not so disastrous a thing with the ball bearing if it does happen as with any other type of bearing.

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*Mounting and Maintenance of Roller Bearings on Electric Motors**

By D. E. BATESOLE†

MR. Chairman and Gentlemen: Before I read my paper it might be well to explain briefly the position of the Norma Company of America, which I represent, with respect to roller as well as ball bearings. For many years, as many of you know, we have been engaged in the manufacture of small ball bearings. A few years ago we became associated with the Hoffmann Manufacturing Company, Ltd., of Chelmsford, England, one of the pioneers in the development of the anti-friction bearing. They have been manufacturing roller bearings for a number of years past and have been mounting them very extensively in electric motors both in England and in Canada. The general character of the Hoffmann roller bearing is shown by Fig. 1.

In making up this paper I have carefully looked into their standard practice and find in general that the methods used in the mounting and protection of the bearings do not differ materially from those in this country.

The subject of mounting and maintenance of ball and roller bearings, wherever applied, is one which is of supreme importance and I believe that I express the gratification of all makers of anti-friction bearings that this subject is receiving the attention which it properly deserves. Every user of ball or roller bearings, I believe, has had in the past instances where the bearings have been improperly mounted or improperly taken care of, with resultant unsatisfactory service. The selection of the proper size of bearing for a particular job is a comparatively simple one when the loads are known. The use of conservative load tables and the usual practice of selecting bearings several sizes larger than the calculated loads in most cases insures the use of a bearing sufficiently large. But no matter how large the bearing selected, improper mounting and improper care may cause it to ultimately give trouble. A careful study of this subject is, therefore, welcome by both the user as well as the manufacturer of anti-friction bearings.

In general, similar methods of mounting and care are required for the roller bearing as for the ball bearing. Both are made to high precision with close internal tolerances so that careless methods used in handling them will be equally as destructive. The conception of the term "roller bearing" has come to mean something quite different in the past few years than it did in the early days when "roller bearings" were very crude indeed. Many of you will recall the time when ball bearings were in their infancy and you will recall the troubles encountered in using them due to their crude design and workmanship. In the same way that the ball bearing has had to live down its old reputation so has the roller bearing had to dispel the old idea of a crude assembly of parts which passed under the name of "roller bearing." In the old days, a few pieces of unfinished cold rolled stock running on an unfinished shaft constituted a roller bearing. Now, however, we have "precision" roller bearings in the full sense of the word. Rollers are no longer left unfinished, but are ground and lapped to the ten thousandth part of an inch and races made to a like precision. Not only that, but they are made of special alloy steel, uniformly hardened throughout, to give the greatest toughness and hardness to both rollers and raceways. The rollers which are here for your inspection are samples of the latest development in the art of roller making. They are standard rollers used in the makeup of the complete roller bearing. It will be seen, therefore, that the "precision" roller bearing of the present day is an article fully the equal of the present day ball bearing in quality and workmanship.

FIG. 1.

It might be well to pause here for a moment to point out the accuracy to which the rollers which you have found before you this evening are made. The Hoffmann roller is square in dimension; in other words, it has the same length as diameter. The sam-

*Presented at Chicago, January 11, 1924.

†Mechanical Engineer, The Norma Company of America.

ples which you have are respectively ⅜-in. and ¼-in. rollers. They are held to .0001 in. to standard in diameter and .0002 in. to standard in length. The rollers may be used for checking micrometers, gauges, etc., where the accuracy is not to be held closer than .0001 in. or .0002 in.

In order to discuss intelligently the mounting and care of roller bearings, it is well to consider the various types which are used, since methods vary to some extent depending upon the type of bearing. Roller bearings may be roughly divided into three types: Tapered roller bearings, straight or cylindrical roller bearings, and special roller bearings having neither straight nor tapered rollers. (See Fig. 2.) Of the last type, there are a great number, some having rollers of concave and some of convex outline. Usually these types have been developed to give them a self-aligning feature. They, however, have not yet come into very general use and the difficulty in manufacturing them to sufficiently accurate limits has made their cost almost prohibitive, except on very large work.

Tapered roller bearings are designed for the purpose of carrying end thrust as well as radial loads. It is necessary to adjust them endwise to eliminate all radial play. For this reason they are not well suited for use on electric motors. If the bearings are

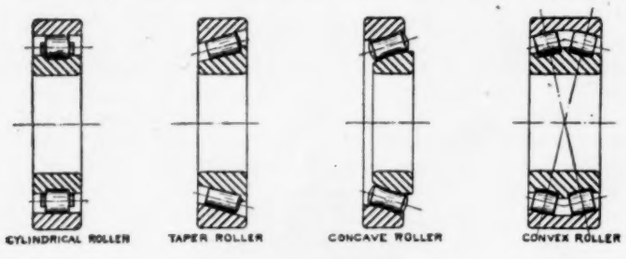

Fig. 2

adjusted too tightly there is danger of cramping them when the armature shaft expands due to heating. On the other hand, if adjusted too loosely there is naturally some radial play. The best results with this type of bearing have been on very large motors where two roller bearings have been used at each end, either with the open sides mounted toward each other or with the open sides outward. In this way expansion or contraction of the armature shaft is taken care of, since one pair of bearings at one end of the motor may be locked in position to locate the shaft and the other pair allowed to float endwise. This arrangement is quite similar to the usual practice where ball bearings are used.

Straight or cylindrical roller bearings may be subdivided into two classes: Those having short rollers and those having long rollers. Short rollers are those having a length approximately equal to their diameter. The long rollers have the advantage that they more nearly approximate the proportions of the sleeve bearing and, therefore, very often replace sleeve bearings without many changes being made in the motor end bells. They offer the possibility of much trouble, however, if the axes of the rollers do not remain parallel with the shaft axis. They have a tendency to skew around so as to not roll with the full length of the roller, bearing against the inner race and they very quickly become worn at the ends. To overcome this objection roller bearings are now made with flexible long rollers and they are quite successful for certain work and at certain speeds.

The short roller bearing approximates the proportions of ball bearing and many of them are exactly interchangeable in size with the standard sizes of ball bearings. This makes it possible to replace a ball bearing with the same size of roller bearing if the ball

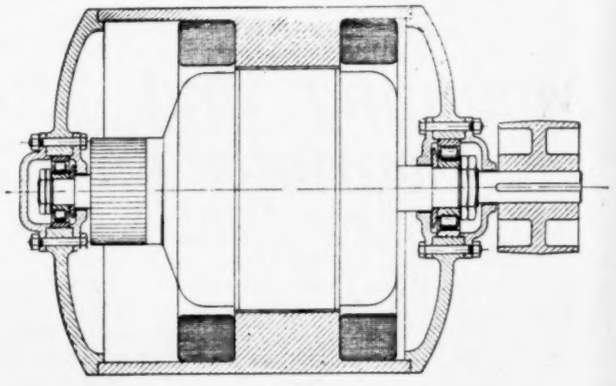

Fig. 3

bearing does not give satisfactory service due to being overloaded. The short rollers do not offer the same possibilities of trouble as the long rollers if they are out of line with the axis of the shaft, because they are not so apt to chip off and wear at the ends.

The straight roller bearing has the disadvantage that it does not carry much end thrust. Certain types are made with the rollers designed to carry the thrust against their ends. (See Fig. 4.) If the rollers are lapped on their ends as well as the sides of the channels in which they roll, they are very satisfactory if end thrust is not large. In ordinary motor service the end thrust is not usually very high, but if it is high, it is the best practice to provide some means other than in the bearing itself to take care of this thrust. Some successful applications have employed friction surfaces; in other words, the end thrust is taken by one surface bearing against another. (See

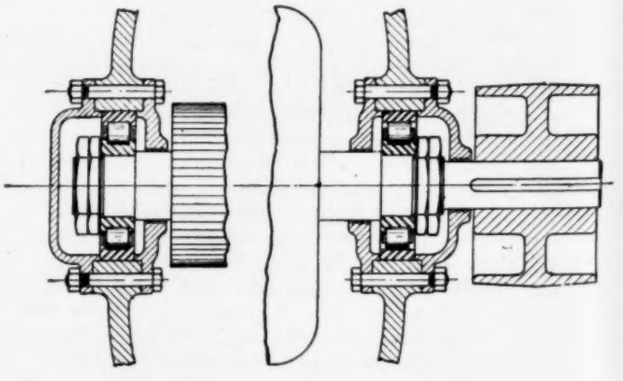

Fig. 4

Fig. 3.) If well lubricated and at relatively slow speeds, this system is quite satisfactory.

Where the speed is high and the end thrust not too great, ball bearings of the deep groove radial type have been used in conjunction with the roller bearing. In this case the ball bearing is used to carry end thrust load only and the roller bearing is used to carry radial

load only. However, where the end thrust is unusually heavy, such as for example where there is a worm drive, this thrust load should be carried on a ball thrust bearing.

In our experience, the best type of mounting for motors up to 40 or 50 hp. is the combination of a ball and a roller bearing, as mentioned above, one at each

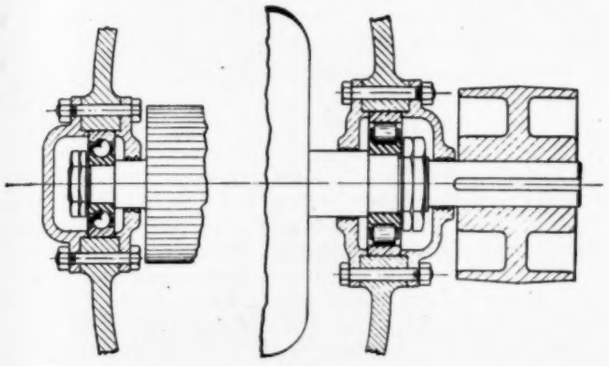

Fig. 5

end of the motor. (See Fig. 5.) The roller bearing is used at the pulley or drive end of the motor and the ball bearing at the opposite end. The ball bearing is locked in position endwise and is used to position the armature. It carries both the radial load, which is present at the closed end of the motor, as well as whatever end thrust may be present in either direction. The roller bearing at the drive end of the motor takes no thrust and has its entire capacity available to take care of radial loading. Furthermore, it allows expansion of the shaft since it is free to move or adjust itself endwise without hindrance. In many cases the same size of roller bearing may be used at the drive end as the ball bearing used at the other end, thereby permitting of interchangeable motor end bells.

For motors over 50 hp. we believe it to be the best practice to use a roller bearing at each end of the motor with a ball bearing at one end only to take care of the end thrust. (See Fig. 6.) In this case, the housing should be cut away slightly around the outer

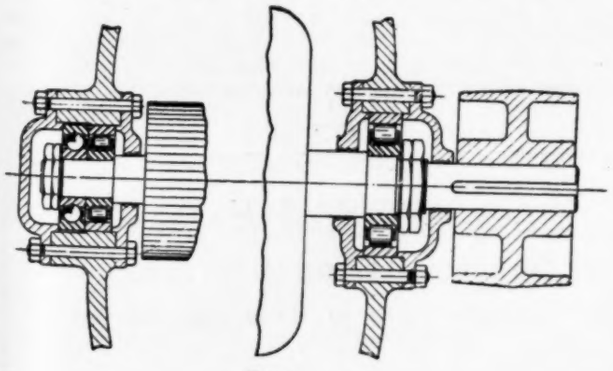

Fig. 6

ring of the ball bearing to prevent it from carrying any of the radial load. The mounting arrangements as mentioned here are, of course, for motors in general service work. Where motors are used for certain special purposes it is sometimes necessary to study the conditions carefully and recommend special mountings to suit them

Where there is an exceptionally heavy end thrust it is advisable to provide a ball thrust bearing to take care of this thrust. (See Fig. 7.) If the thrust is always in one direction a single direction ball thrust bearing will suffice, but if it is at all likely to be in either direction, a double direction ball thrust bearing should be provided.

Coming now to the actual mounting of the roller bearing in the motor, there are three main points to be considered. First, the fits to be used. Second, the protection of the bearing during assembly and after it is in service. And third, the lubrication of the bearing.

The fit allowances used in mounting roller bearings to the shaft do not vary considerably from those used in the mounting of ball bearings. A light press fit is recommended for this purpose. The inner ring should be pressed home against a shoulder on the shaft or against a bushing or some other locating surface. Preferably a nut should be used to clamp the ring securely against this positioning shoulder. There is some disagreement on this point among motor manufacturers, but it is really, the only way to insure that the inner ring is to be held firmly in position. A

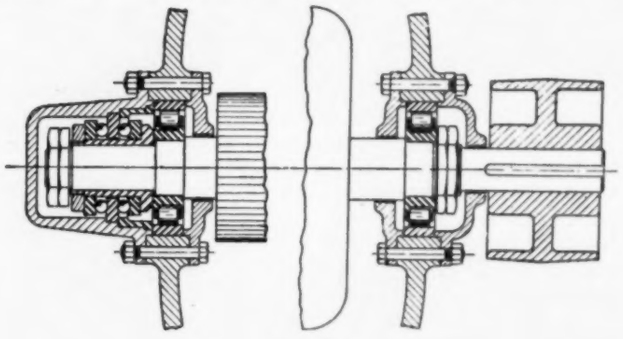

Fig. 7

locking device should, of course, be provided to see that this nut does not become loosened by vibration.

The practice used in mounting of roller bearing outer rings sometimes varies from ball bearing practice. If the roller bearing is a complete inseparable unit like a ball bearing, then approximately the same fits are recommended as used with ball bearings. However, a separate outer ring permits of a more snug fit of this ring in the housing since it can be mounted separately. It is the usual ball bearing practice to allow enough clearance between the outer ring and its housing to permit the ring to slowly creep around. With a separately mounted roller bearing outer ring, it should be mounted snugly and, furthermore, should be clamped endwise. In this way there is no chance for looseness and peening out of the housing to develop. The fit allowances of course vary with the size of bearing and it is good practice to follow as nearly as possible the recommendations of the bearing manufacturer in each case.

The need of protecting the bearings from the entrance of dirt, chips, water, etc., cannot be over-emphasized. As mentioned before, anti-friction bearings, either ball or roller, are made with very small internal clearances and it is absolutely necessary to keep all foreign matter out of them. In the majority of cases, careless handling of the bearings during assembly, rather than poor protection after they are mounted,

is the cause of failure. Bearings are usually shipped covered with some slushing material to prevent rust. This material has a great affinity for dirt and chips and anything coming in contact with it is readily picked up. So it is desirable to keep the bearings in their original packages until ready for use and then to see that they are not laid where there are steel chips and filings to be gathered up with them. In this connection, it is highly recommended that wherever possible some one person should be designated to install the bearings in the motors and no one else should be allowed to handle them. This person will soon become quite expert at this work and much better results will be obtained than if every workman in the plant is allowed to handle them.

In many cases, bearings after being washed out, are subjected to an air blast to rid them of all particles which might adhere to them. Many times this air is full of dust particles as well as water, and this is very injurious to the bearings. Extreme care should, therefore, be taken when it is necessary to use the air blast for this purpose.

The protection of the bearing after it is assembled is also very important. In general, devices used for this purpose likewise serve to retain the lubricant in the bearing housing. Practice varies considerably among users of roller bearings in this regard. Close running clearances maintained between the shaft and that part of the housing through which it projects are very effective. Also the use of interlocking projections is very good. This is sometimes called the labyrinth system of protection where foreign matter must pass around a number of corners before entering the bearing housing. Where grease is used as a lubricant, small grooves in the bore of the housing immediately surrounding the shaft are very satisfactory, since the grease fills these grooves and prevents the dirt from entering. Felt is sometimes used, but unless properly mounted is very apt to become ineffective after a short time. It is very difficult to describe a type of protection which will suffice in all cases, since conditions of operation usually demand certain special types of protecting devices. It is well where a motor is to be operated under unusual conditions, such as conditions of high temperature, excessive moisture, etc., to submit the problem to the bearing manufacturer who has no doubt had experience with his bearings under similar conditions.

The problem of lubrication is one which has perhaps received more attention than any other point in the use of anti-friction bearings. The question of grease instead of oil or oil instead of grease has been discussed many times. Here, again, conditions of operation will determine the best lubricant to be used. With roller bearings in ordinary motor work we believe grease to be the best lubricant. It does not leak out of the housing as easily as oil. This is very important in electric motors where the armature and field windings should be kept free from oil.

A pure mineral grease perfectly neutral and of about the consistency of vaseline, is the most suitable for roller bearings. It should not be of such a nature as to readily absorb moisture and should contain no solidifying matter, such as chalk, graphite, etc., which is likely to clog the bearing or cause wear.

It is recommended that the housing be crammed with grease when the bearing is set to work. The rollers will free themselves in the first few revolutions, but the grease will cling to the surfaces not actually in contact and protect them from rust. It will be found that one such charge of lubricant will last for a very long period. When it is necessary to add additional lubricant care should be taken to see that the grease is clean and free from particles of dirt. Here again, it is advisable to employ some one person, if possible, to attend to this work so that it may be done intelligently.

It may appear from what has gone before that the satisfactory working of a roller bearing is a particularly difficult thing to attain, but such is not the case. With ordinary care in the mounting of the bearing and proper provision for lubrication at regular intervals, very satisfactory service will be obtained over many years even under adverse conditions. As a matter of fact, the roller bearing will stand more rough usage than the ball bearing. This is on account of its heavier load capacity and greater area in contact between the rollers and their raceways. The roller when coming in contact with a chip or particle of dirt will perhaps receive an indentation on its surface, but there is still a considerable area which will bridge the small gap thus caused. In the case of the ball bearing, when a ball once receives an indentation on its surface, it will eventually cause the destruction of the whole bearing. This should not be taken to mean that slipshod methods of mounting may be used with the roller bearing. On the contrary, it is just as advisable to use careful methods in the mounting of roller bearings as in the mounting of ball bearings.

In conclusion, I would like to suggest that an attempt be made to furnish the members of the A. I. & S. E. E. with a standardized recommendation for the mounting of both ball and roller bearings to mill type motors. It may be possible, however, that the Association already has a committee for this purpose, but if not, it seems that it would be well to appoint such a committee. This work could include recommendations on fit allowances for various sizes of bearings, locking devices, protecting devices as well as standardized mounting arrangements. While there is a considerable difference of opinion in regard to the above points, I believe that some definite recommendations could be arrived at without much trouble and I am sure that such recommendations would be welcomed by the men of the Association as well as by the bearing manufacturers themselves.

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Ball Bearing Applications and Maintenance

By H. R. REYNOLDS*

MR. Chairman, Mr. Batesole's talk of roller bearings, because they are interchangeable with ball bearings, must of necessity have covered some of the points which I will also bring out. The object of the talk, as I wish to present it, is to explain to the gentlemen present how bearings should be selected, and how mounted. The slides will not be used until after the talk, and then I will give a word of explanation to show just how they bring out the points which I make as I go along.

It is common practice for most of the ball bearing people to state in their catalogues that all cases of engineering questions should be referred to the home office. Ball bearing practice has advanced to a point where I think it is possible to give you formulae which will give you some conception of what your ball bearing should carry.

Understand that the talk tonight is on bearing size and selection, as well as mounting. No talk of bearing sizes is complete without the description, first of the fact that ball bearing makers list their bearings in three series. I am talking strictly of the single row radial bearing.

These three series are commonly known by the numbers, 200, 300, 400. Two hundred is the light, three hundred the so-called medium, and the four hundred series is the heavy.

The 200 series for industrial application by the Fafnir Bearing Company is very seldom recommended, and then only where shaft sizes, because of design conditions, are large in proportion to loads involved.

The three hundred-series are in industrial applications very much favored by us. In most cases these bearings are larger than the requirements, giving the necessary factor of safety which is needed in all good engineering.

The 400 series are used where short shafts and very concentrated loads are in the design, and the need of the trade for the 400 series is very seldom met with. In other words, it is the one series we make the fewest of.

In explanation of the 200, 300 and 400 series, the 208 bearing, 308, and 408, would carry the same bore diameter, but a 208 carries a ⅜ in. ball; a 308 would carry a 9/16 in. ball, and the 408 would carry a 13/16 in. ball, showing the relative capacity of the bearing changes very rapidly in the three series.

Another factor of satisfactory service, aside from ball sizes, is revolutions. In the automotive field, 40,000 miles would probably be considered a good long life for a ball bearing. In the industrial field, in electric motors, especially, the same number of revolutions would only be three or four months' service. So you can easily appreciate that a company manufacturing ball bearings finds the industrial field the hardest one to satisfactorily design ball bearings for, and the best bearings we make are saved for industrial applications. Consequently, because of the difference in the service, the 200 series are largely used in the automotive field, and most of the designers work the 300 series ball bearing into what we call the industrial field. They are represented by all types of machinery and electric motors.

The biggest factor in the satisfactory use of bearings is keeping them clean. That has already been presented. I will emphasize it. A ball bearing primarily is a pair of rings, an outside ring and an inside ring, and a set of balls. These balls run in fairly deep grooves, and any foreign matter getting into the bearing is grit to the bearing, even if it is what they call "fly" in the textile mill, the lint that floats around in the air. That is a foreign element, and the balls in the bearing have to climb over it, or flatten it out some way or other. It takes up space. The housings must be so designed as to exclude foreign material.

Another factor in the grit proposition comes in in bearing selection. In a cement mill, where grit is bound to get in on grinding operations, where there is considerable emery and hard steel to be considered, the closure might be very effectively designed to keep out nearly all of the dirt, but the man who lubricates these bearings will invariably wash in a small amount of it with his lubricant, especially if he uses oil.

Under these conditions, we believe in using the bearing with the larger ball. It is not a case of carrying capacity, but a case of area. It varies as the square of the diameter, so you get a tremendous lease of life by using a little larger ball.

Right here I will mention a very characteristic condition of ball bearings that wear out from dirt. Anybody that has had experience with ball bearings recognizes the sloppy bearing, the one that has been in service about as long as it can be, and has gotten loose and sloppy. If there are no checks in the ball raceways and the balls are not flaked they have been lapped out by the dirt that has got in through carelessness or through improperly designed housings. A bearing that fails through fractured raceways, flaking of the raceways and of the balls, generally fails from overload.

Overload may be caused by an initial load beyond the designer's conception, by the mounting of the bearing, and by foreign elements getting between the balls and the races.

Selection of bearings, that is, the load rating of bearings, has been in the past very much tied up to the Stribeck formula, which is $\frac{Knd^2}{5}$. This gives the answer in kilograms. To reduce or change to pounds, roughly, it would be multiplied by two and a half. The factor K is variable with the speed. N is the

*Engineer, Fafnir Bearing Company.

number of revolutions. D is the ball diameter in eighths of an inch, and 5, a factor put in to take account of the fact that only part of the balls are loaded. Because the factor "K" varies with the speed it makes this formula uncertain and so prevents its universal use.

In commercial recommendations the Fafnir Company hold very closely to Professor Goodman's formula. The factors in the Goodman formula are known, and those that are not known can be easily calculated, especially if the customer has one of the company's bearings at hand.

The Goodman formula is worked out only for the deep groove radial bearing. Especially the alloy steel, high carbon, high chrome type. Don't try to apply it to any other. That Goodman formula I will put on the screen a little later and anybody that wishes it can copy it down.

In emphasis of the previous speaker's statement, even though you have the formula and the closures have been shown you, and you feel quite competent that you know it all, in any new application of bearings to motors, or any other unit, through the large experience which the bearing companies necessarily have, it is to your own advantage, as well as theirs, to take up with their engineering departments your problem, and get the benefit of their advice. This requires complete co-operation on the part of the possible user. We must know all sources of load, and sometimes we will write and ask you for a few more you have overlooked. Be patient with us. We are trying hard to design perfect and satisfactory service into your machine.

Another thing which I will re-emphasize is the training of the mechanics who are to mount the bearings. The mounting of ball bearings sounds much worse than it really is. It is like your troubles. You generally see the butt end of them first. After you get over the butt you find it is much simpler than it looked, especially a year or two later, when you brag about them.

In aid to the mechanic who is mounting the bearings most companies publish a set of fits, such and such a diameter for such a size bearing, if the shaft revolves, and another set if the housing is revolving. These fits and the commercial limits which we work to, these limits decided by the Society of Automotive Engineers, present two extremes. If the extremes meet, either the bearing will be too loose or too tight.

We prefer that a customer who is using bearings in quantity, and whose unit is tied up in the service of the bearing as much as any other one element, make what we call a selective assembly. In other words, if a bearing is too loose, his mechanic should take that one off and put another one on, because there is bound to be a shaft a little later which is too tight, which will take that bearing.

Another thing about bearing application which can't be too strongly emphasized is the fact that for capacity a ball bearing, and if our friend's roller bearings are of greater capacity than a ball bearing, those roller bearings, too, are narrow in proportion to their capacity, and if they are not carefully fitted to the shaft, due to the loads which they are called upon to support, and the small amount of shaft area in contact with the bearing in the ring, the shaft will peen under the bearing; especially is this true in motors belt-connected, where the belt is laced instead of being cemented. The blow of the belt lacing, some of the commercial types, especially, setting up a dynamic shock which will sometimes even when carefully fitted, peen the shaft under the bearing. That is one of the reasons why we say, make the bearing a close fit to the shaft.

Another feature which I will bring out here, not quite in line with the subject, but that is the fit-up of the ball bearing. A number of customers, one customer thinking one thing, and another customer thinking another, are continually dictating to us as to what clearance we shall have between our balls and races.

Now, a ball bearing properly fitted up is quite loose. It has to have a certain amount of internal clearance after it is mounted, and as my friend from the Norma Company said, there is an expansion of the inner ring, due to the press fit. We like to allow for that expansion in the fitting up of our bearings, but if a customer insists on having them tight he is going to get initially an overloaded bearing from the expansion of the inner ring.

Another point to bring out here is the shoulder on the shaft. The ball bearing, as I said before, is narrow, so narrow that driving it on, unless it is put on under an arbor press, and everything is kept square—it can be canted on the shaft. The shoulder corrects this condition. If it is driven solid against a carefully turned shoulder, which is square with the axis, it will come back to line. If the shoulder is not high enough, the little round corner on the inner ring of the bearing will bear against the shoulder. This corner is not finished true with the rest of the bearing. It is an unground surface. It is put on there so that a small corner fillet may be put on the shaft to strengthen it at this point. The shoulder must always come well above the round on the ring. The shoulder has two functions; to square up the bearing and furnish an abutting shoulder to lock the bearing against. The lock nut described by the Norma gentleman, which they use, being a slotted nut with a lock screw in it. We use it on what we call a taper sleeve bearing. We find that it is better to supplement that with a keyed washer between the nut and the bearing in the ring, this keyed washer being keyed to the shaft, preventing a slip between the two surfaces, the fault of this slotted nut lying in the fact that the mechanics don't always bind the adjusting screw tight enough to deflect the main threads.

The best way to mount ball bearings, especially if the production of the units which use them is large enough, is to have an oil tank, the temperature of the oil to be from 250 to 300 deg. The bearings should be left in it long enough to come up to heat. That only takes a minute or so. In a progressive assembly one lot can go in as the others come out. It will expand the bearings beyond the fits which are given in the ball bearing catalogues, so they can be dropped down against the shoulder, and they will come down nice and square. Everything is assured, and as they cool off and shrink back they will be on the shaft very nicely.

If they are driven on the shaft, especially if not driven straight, they are bound to shear the metal

a little; more so if too tight than a good fit. That shearing means instead of a fit to start with, you have something less, and that metal generally packs in between the face of the bearing and shoulder on shaft and causes a little bearing cant which is somewhat destructive.

The closure description I will leave until we show the lantern slides. The closures are, in my opinion, a question entirely of conditions. If a man is building electric motors, he is going to send his motors to all places, and some type of closure which would be effective in a cement mill and also a flour mill — one place where you have to keep out grit and another place where you have to keep in grease—if they are built to do both things, are quite expensive.

The standard closure used by motors, ball bearing equipped, are simply two small felt grooves. I agree with the Norma gentleman in condemning, more or less, the felts. We find—this may sound as though I was talking up some company's product, but I am only trying to make my point clear — that the Goodyear Spiral Steam Packing is about as good a packing for the felt grooves as anything we have found. It does not seem to score the shaft, does not carbonize, and it makes a very effective seal in the lubrication of ball bearings.

Our most recent experiments have led us to the conclusion that, provided the proper grade of grease is selected—grease is by far superior to oil up to speeds as high as 10,000 revolutions per minute. We have numerous satisfactory applications in the woodworking field, where the speeds run around 7500, grease packed, the grease being applied once in three months, and it would not need to be applied so often, if we could get a grease which would stay a grease, but grease that is satisfactory in one barrel is not satisfactory in the next. I would be glad to argue that out with any oil men here. I could tell him of a good many instances. I think all the ball bearing gentlemen would bear me out and he'd have quite a talk on his hands if alone.

We have a lot of trouble in getting a consistent grease. Most of them, from one lot to the other, will vary enough to condemn one or the other of the lots. We recommend a grease a little stiffer than vaseline, a mineral grease, and a neutral grease, with a metling point as high, at least, as 150 deg.

The frequency of renewal depends entirely upon the type of grease you use. We have run ball bearings in our own plant for six years with one application of grease, and have found grease in the housing when we took them down. We have run ball bearings in our plant for six monhts and found what looked like hard soap in the housing and nothing else when we took them apart, and it was out of another barrel of the same grade from the same company. So we can't recommend the grease. We can tell you what we are using today, and we recommend you use it often, so as to keep it soft in the housing.

We will now put on the slides.

One of us thinks he has the best separator and another thinks he has the best balls and the next man thinks he has the best material—taking it altogether they are about the highest developed, specialized unit which is available as a bearing.

Fig. 1. This picture shows an electric motor, as they are ordinarily mounted. You will notice here the lock nut, showing the wire seal, the high shoulder, the point where the inner ring fits the shaft, and something which I did not mention, the locking of one outer ring, and the clearance each side of the other. That is one of the main features of a ball bearing application, being a single unit in itself, and having in proportion to the radial capacity a high thrust component—for a majority of cases it is an entirely complete bearing, both thrusts and radial loads being taken care of by the single row radial type.

Fig. 2. This is a large section showing the common practice in motors of boring the end bell straight through and using two caps. The photographer took this picture upside down. It is supposed to show a vertical application. If you can imagine this just turned over you will see here a cylindrical seal, forming an oil well. The depth of that oil must be up to this little flipper on a sheet metal disc, just under the bearing, between that and the shoulder, at slow speeds, and this little flipper here — and it does not have to be very big — will pick the oil up and sling it into the bearing. It tends to drop back again and is picked up again and is a very effective means of using oil on a vertical mounting.

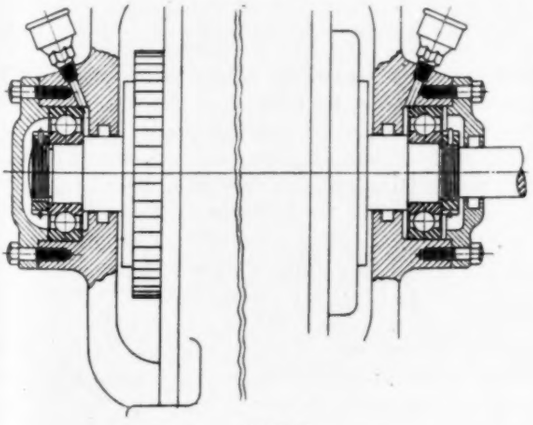

FIG. 1.

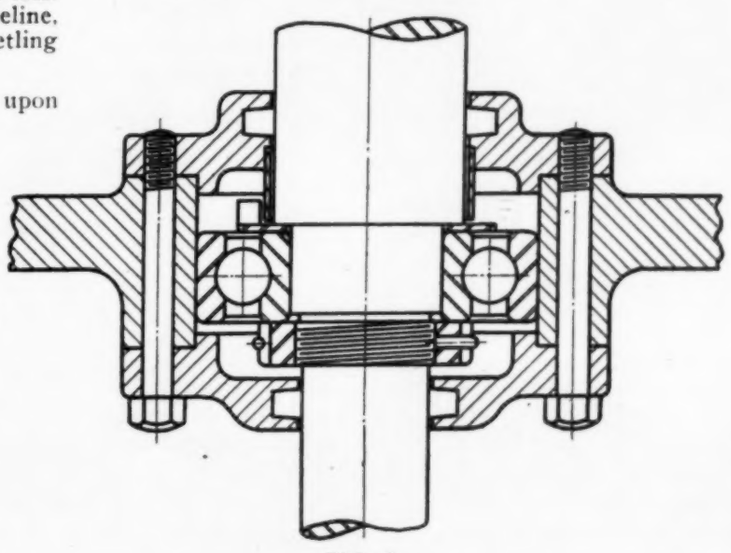

FIG. 2.

A grease packed housing, if the grease is as dense as you can use satisfactorily at normal speeds, will leak very little grease. One of the biggest advantages of grease is the fact that it does not work out of the housing. It must have, however, each side of the balls, considerable space to absorb the motion im-

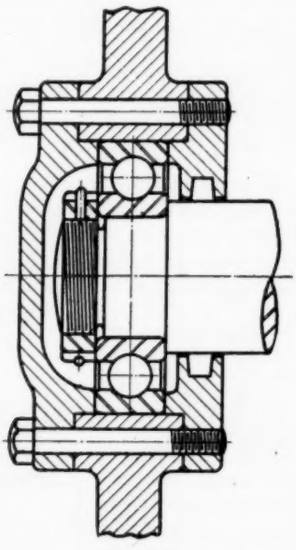

FIG. 3.

parted to it by the balls and the separators traveling through it.

A majority of ball bearing people build a separator that follows the contour of the ball—it is like pushing so many little paddle wheels through the grease. If your housings are too close to the bearing that mo-

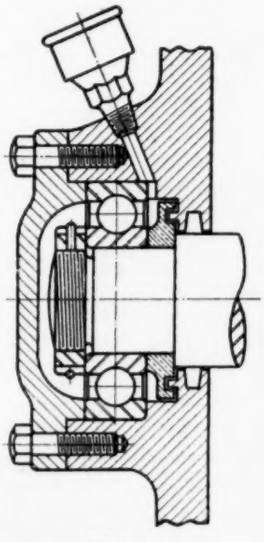

FIG. 4.

tion will force the grease slowly through the end covers—if the grease is too thin it will heat.

In using oil for a lubricant, if the oil is up too high it will set up an internal friction between the balls and the oil, which has been known to even strip the separator from the balls.

Fig. 3. This shows similar characteristics to the vertical one, which I just showed, especially the use of two covers, which simplifies the operation of boring the housings.

Fig 4. This illustrates a very commonly recommended grease slinger. It is very effective in keeping grease in the housing, and is used considerably by the Fafnir Bearing Company in recommendations for applications on machinery manufacturing food products and in textile mills. Unfortunately, it does not combine with keeping the grease in, the functions of keeping the dirt out. If the dirt gets by the point here it simply pushes it into the bearing, so the opposite type of enclosure is recommended by us for applications such as wet grinding, cement mills and similar practice.

Fig. 5. This is a very badly light struck slide. You see the same lubricant seal on the outside, and these little grooves here, we don't felt pack, we just simply depend on lubrication getting in there and making a grease seal that, in connection with the outside slinger, has been used by us very successfully on high speed wet grinding spindles. We get anywhere from a year to two years' service on internal grinding with this enclosure.

This is the formula very carefully worked out by Professor Goodman for the selection of deep groove high grade alloy steel single row radial bearings.

$$\text{Goodman formula} = Knd^3/(RD+Cd)$$
$$K = 2{,}500{,}000$$
$$C = 2{,}000$$

You will notice that I have explained the factors. K and C are empirical factors. N is the num-

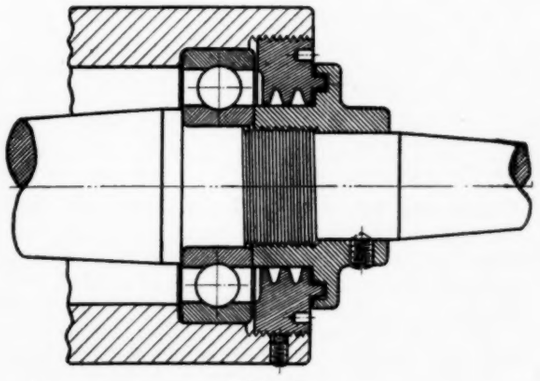

FIG. 5.

ber of balls. D is the ball diameter, not an eighth of an inch, but the actual diameter. R is the speed in revolutions per minute. Big D is the diameter of the outer ring ball raceway over the balls. If you know the size of the ball and bearing you can easily calculate this factor. This formula we use for our bearings direct when the loads in industrial applications are intermittent. For continuous service we use a factor of one and a half, and I can assure you gentlemen that excepting where cases are unusual that a standard high grade bearing will give you a conservative rating if you use the Goodman formula. Our laboratory experiments have worked that out on graph after graph, taking several bearings in a series, loading them and running them.

Application of Anti-Friction Bearings

By H. E. BRUNNER†

THE purpose of this paper is to consider the relation which the anti-friction bearing has to the design and operation of electric motors. This will be discussed with particular reference to conditions of operation and maintenance.

It is now generally recognized that anti-friction bearings constitute an important factor in the motor industry. The trend of progress and its insistent demand for increased efficiency of production, is largely responsible for the present development of such equipment. The same compelling influence gives assurance that the future will see an ever increasing progress in utilizing the possibilities of anti-friction bearings. It is quite generally conceded that they afford a sure means of obtaining increased economy in the operation and maintenance of electrical equipment.

The history of the development of this phase of the motor industry is typical, and its milestones of progress are comparable with those marking the introduction of ball bearings in other fields. The first experimental work was handicapped by the limited experience of both the motor designer and the anti-friction bearing engineer. The result of their effort embodied in the first commercial applications was confronted in operation with that innate prejudice which awaits any innovation that is a marked departure from the beaten paths of practice. It is, therefore, not very surprising that the early attempts to build ball bearing motors were acclaimed as only a mediocre success.

The commercial development of electrical machinery in both design and construction, has been one of the outstanding accomplishments of the last two decades. The achievements of this period are a lasting tribute to the skill and ingenuity of the pioneers in this industry. Closely paralleling their progress there has been an equal, although possibly less spectacular, development in the art of manufacturing and utilizing anti-friction bearings. Today, the experience of the electrical engineer, in collaboration with that of the engineer who has specialized on anti-friction bearings, makes it possible to design a ball bearing motor with definite assurance as to the results which will be obtained in service.

The truth of this last statement is attested by the many installations in successful operation. However, it would be absurd to infer that the entire problem of applying anti-friction bearings to electric motors, has been solved in all its involved phases. This is no more true in the case of ball bearings than with plain bearings. It can readily be understood that the entire field of motor manufacturers with their widely diversified products, would undoubtedly entertain numerous individual view-points. The same is true of ball bearing manufacturers, although it can be safely said that this exists to a lesser degree. Such conditions are nothing more than an indication of an essentially normal and healthy state in a progressive industry. It is by no means necessary that a unanimous opinion exists in the analysis of detail, provided the broad lines and fundamental principles can be agreed upon and accepted. We believe that such an agreement is about to crystallize into an accomplished fact, due in no small degree to the insistent efforts of this association.

The individual viewpoints of the motor manufacturer has been cited as a desirable, progressive influence. There is, however, a limit to the extent of its usefulness particularly when the industry can be said to have merged from a transient stage of development into a stabilized state of accomplishment. Under the latter condition there exists a danger that the user of the product may be penalized by a failure to appreciate his individual needs and requirements. Accordingly, there is a definite demand for the establishment of certain stands, giving comprehensive recognition to the requirements of service and at the same time providing as free a hand as possible for the ingenuity of the designer. Exactly such is the nature and purpose of the anti-friction bearing standards which are being proposed for electric motors. Hand in hand with this standard of bearing capacity there is a need for agreement as to the fundamental requirements of mounting design. In this way only will it be possible to realize the full benefits that may be obtained.

Relation of the Bearing to Design.

The inherent characteristics of anti-friction bearings require that due consideration be given their peculiarities, in order to fully utilize their advantages. This means that in applying them to an electric motor the mounting should conform with the requirements of the bearing. The method of securing the bearing to the shaft, the housing design, and the provision for lubrication are all of primary importance. In addition the procedure for assembly and disassembly of the bearings in the motor, must conform with the individual condition and established practices. An attempt will be made to indicate the relation of these factors to the application of anti-friction bearings in general purpose motors, endeavoring to demonstrate the underlying principles rather than to enlarge on detailed methods.

Shaft Fit.

Anti-friction bearings have been mentioned as having certain peculiar characteristics. These are more apparent by contrast with the usual type of plain ring oiled bearing. The latter provides for supporting the journal load on a film of lubricant interposed between the shaft and bearing surface. The physical properties of this film limit the allowable pressure which can be sustained without its rupture, and consequent seizing of the bearing. This pressure together with the journal diameter establishes the lenght or width of bearing required by a given load at a given peripheral speed of shaft. On the other hand, the anti-friction bearing supports the journal load by the elastic deformation of steel balls or rollers in metal to metal contact with their races. The ball or roller diameter must, of course, be proportioned to the load and speed, but beyond this no particular limitation is imposed on the width of the bearing. As a result the width of a ball or roller bearing is quite small compared with that of a plain bearing. The relatively small projected area between the bore of the inner

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race ring and the outside diameter of the shaft, leads to high values of pressure under the usual operating loads of the bearing. This unit pressure will, under average conditions, amount to from 500 to 1,000 lbs. per square inch.

The intensity of pressure between the bore of the inner race and the outside diameter of the shaft makes necessary a positive means for securing the two together in order to prevent relative rotation. It can readily be seen that if an actual clearance exists between these parts there will be a rotation similar to that of a spur gear meshing with an internal gear. This creeping, under the high unit pressures involved, will rapidly develop destructive wear of the shaft with attendant heating. To overcome this tendency it is necessary to have a driving force in excess of the frictional force due to the unit pressure between the ring and the shaft. This shows at once the desirability of employing a press fit of the inner race upon the shaft, for the dual purpose of removing the clearance and affording a frictional driving force by the elastic expansion of the inner race. A second possibility is offered by clamping the inner race axially between a shoulder on the shaft and a suitable locknut. The usual practice is to provide both a press fit and a locknut, but with the wide series bearings the locknut may be dispensed with. This is due to the greater ring section which decreases the unit pressure and increases the driving force. In such a case the bearing should be located against a shoulder on the shaft and secured axially by a pressed-on sleeve.

The desirable value of press fit for the inner race on the shaft, is limited on the one hand by the minimum required to secure positive driving and on the other by that maximum which results in excessive hoop tension in the race ring. The hoop tension constitutes a secondary stress, superimposed on the ball or roller contact stresses, and will, if excessive, limit the effective load capacity of the bearing. Experience, however, has demonstrated the practicability of controlling the press fit by grinding the bearing seat on the shaft and by utilizing the small tolerance on bore, which is standard for ball bearings. In this way a suitable press fit can be obtained to insure a positive driving security and at the same time limit the hoop tension in the ring to permissible values.

In cases where the bearing size is quite ample for the loads involved and when conditions of assembly impose restrictions on the use of suitable press fits, it is frequently possible to rely on a very light press fit together with a suitable locknut for clamping the inner race. This method lacks the effectiveness of a substantial press fit, inasmuch as it does not remove the radial clearance and the tendency of the ring to rotate on the shaft. Here too, experience has been the best guide in establishing the availability of the method.

Housing Fit.

The fitting of the outer race of the bearing in the end bell of the motor is another important factor in the correct mounting of ball bearings. As in the case of the inner race the unit pressure between the bearing and the housing bore is high, compared with corresponding pressures in plain bearings. This emphasizes the importance of accurate workmanship in the machining operations, to prevent the development of high localized stresses in the support of ring. In addition, the diameter of the housing bore must be uniformly round, so as to avoid elliptical or other distortion of the comparatively flexible race ring.

Mounting the stationary outer race presents a different problem to that of the rotating inner race. There is no necessity for provision against rotation, inasmuch as the tendency is very slight, being that due to the friction within the bearing, only. Actually it is a desirable condition if the outer race does creep

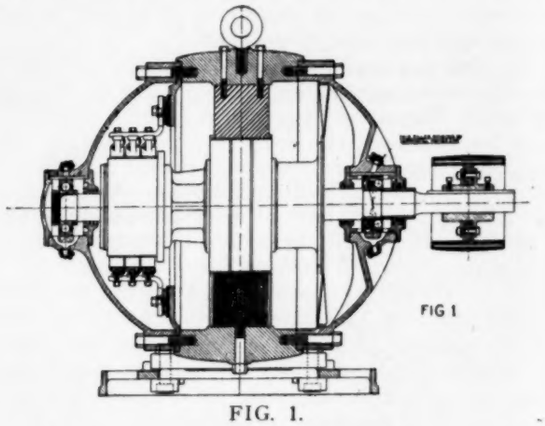


FIG. 1.

from time to time, since in this way the entire periphery of the race groove will be successively subjected to the load. It is thus evident that the outer race of the bearing should be a comparatively free or "sucking" fit rather than a press fit in the housing. The fit is limited on the one hand by sufficient clearance to prevent distortion of the race, and on the other hand by the tendency to excessive vibration, as the result of too great an allowance. As in the case of mounting the inner race on the shaft, the control of housing fit is facilitated by the close tolerances held to by the bearing manufacturer, together with accurate machine work in production of the housings. Experience has shown that it is perfectly feasible to ob-

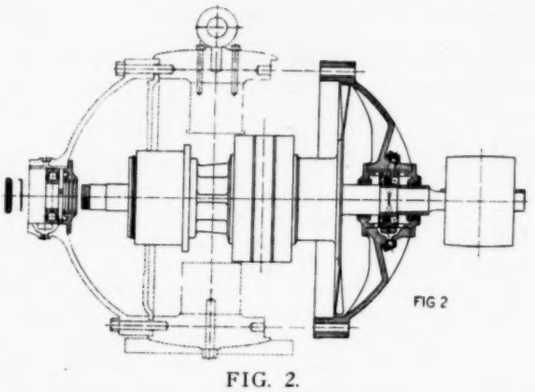


FIG. 2.

serve such tolerances in the machine work as to insure interchanegable parts for quantity production and at the same time provide for the correct fitting of the bearing.

Fig. 1 illustrates a correctly designed application of ball bearing to general purpose motors. Fig. 2 shows the procedure followed in removal of the rotor and Fig. 3, the method of removing the pinion end bearing.

It has appeared desirable to outline above these fundamental features of design which relate to the

mechanically correct mounting of anti-friction bearings. These bear a most important relation to the results which will be obtained from the bearings in service and thus directly affect their maintenance. It follows that in servicing motors and replacing bearings, this same condition of shaft and housing fit should be obtained, if satisfactory performance is to be expected.

Lubrication.

Lubrication is of primary importance in both the design and maintenance of anti-friction bearing motors. The problem of lubrication, as that of mounting, is concerned with the peculiar characteristics of anti-friction bearings, and is essentially different from that encountered with plain bearings. It can safely be said that the whole subject of lubrication of anti-friction bearings has suffered by neglect. This neglect has sprung from the predisposition to eagerly resort to a ball bearing as a sort of "cure all" for the lubrication difficulties of plain bearings. While it is true that the problem is more simple, it is only recently that a full appreciation of its importance has developed, and that a systematic study has been undertaken to determine the underlying principles. In this regard we may

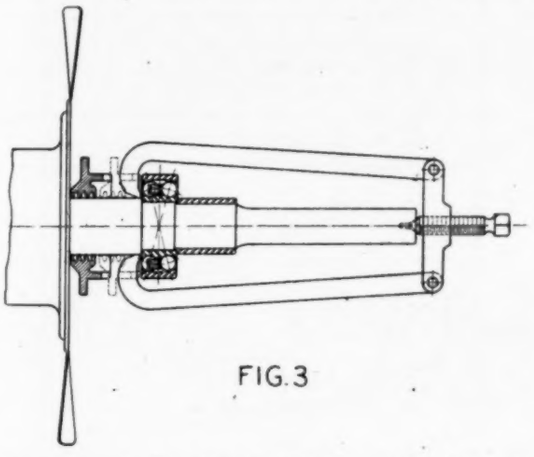


FIG. 3

find some comfort in the fact, that a scientific research into the problems of plain bearings lubrication, is likewise of comparatively recent origin.

Experience has demonstrated that either oil or grease is a suitable lubricant for ball bearings under proper conditions. The choice between these is largely governed by factors, aside from the bearing itself, principally those of design and operation. In electric motors of the general purpose type, grease is to be preferred, as this undoubtedly simplifies both the problems of design and maintenance.

Satisfactory lubrication of the bearings depends on the selection of the proper type and grade of lubricant together with the provision for applying this to the bearing. An important factor in the selection of grease is to obtain one free from any corrosive agent which will attack the surfaces of the bearing. The melting point of the grease must be high enough so that the temperature obtained under running conditions will not cause the grease to melt and disintegrate. In this connection the research and experience of the various bearing manufacturers will be of very great assistance.

A general purpose motor (see Fig. 1), should be designed with ample lubricant space around the bearing to allow of operation for a period of six months without regreasing. Some form of lubricating fitting to be used in connection with a grease gun is the most effective means of applying the lubricant. This serves as a protection against excessive dirt and provides a convenient control of the quantity of lubricant injected. The fitting should feed to each side of the bearing to insure proper distribution within the housing. Dependence cannot be placed on the grease working its way through the bearing. A drain should be provided for washing out the old grease and the connecting passages should be of as ample proportion as possible in order to facilitate this operation.

The use of grease for lubricating an anti-friction bearing motor requires some definite provision for controlling the quantity applied. In general this must be limited to that amount which is required to supply the available housing space from ¼ to 1/3 full. If a greater amount be used there is immediate danger of developing excessive heat from the churning action of the rotating bearing. Leakage will also take place due to the increase of the effective volume of the grease under this churning action. The necessity of closely regulating the quantity of lubricant, in itself, requires washing out of the housing when regreasing.

Enclosure design is intimately concerned with the method of lubrication. The use of oil requires more

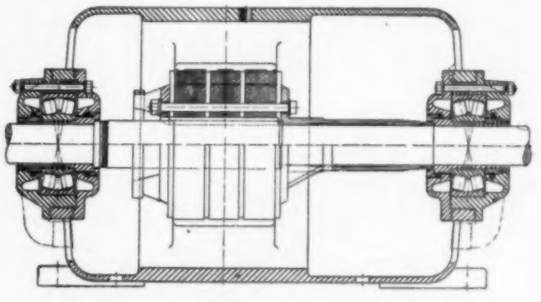


FIG 4

elaborate types of seal between the rotating shaft and the stationary housing in order to prevent leakage along the shaft. It has already been stated that the use of grease simplifies design in this respect. A small running clearance between the shaft and the housing, in conjunction with a series of grease grooves is sufficient to prevent leakage, if a proper grade and quantity of lubricant be employed.

Bearing Capacity.

It is not intended at this time to enter into the problems relating to the correct proportioning of bearing capacity to motor size and type. This subject has been studied in considerable detail for the purpose of arriving at a standardization. It is one which more directly concerns the problem of design, but a few general remarks along this line may be of interest.

A satisfactory basis has been determined upon and the different interests are in agreement, on the matter of adequate bearing size for the standard power ratings of general purpose motors. Such an agreement must, of course, be based primarily on the normal condition of operation. The possibility of providing for the extreme conditions of abnormal service, such as may be encountered in extremely heavy mill drives, should

also be considered. In this respect, the ability of the bearing to sustain the impact of repeated shock loads is of great importance.

A correctly designed roller bearing utilizes a line contact between the rolling elements and provides some means for correcting the tendency of the rollers to skew. The extensive contact thus obtained, as the elastic rollers deform under load, is of value in providing for shock loads. The volume of the material in stress at the contact directly affects the resiliency of the bearing and its capacity to sustain shock loads. In general a roller bearing is superior to a ball bearing in this respect. However, the necessity for alignment of the roller with the axis of rotation is of prime importance, since actual line contact is entirely dependent on this condition. The ball bearing on the other

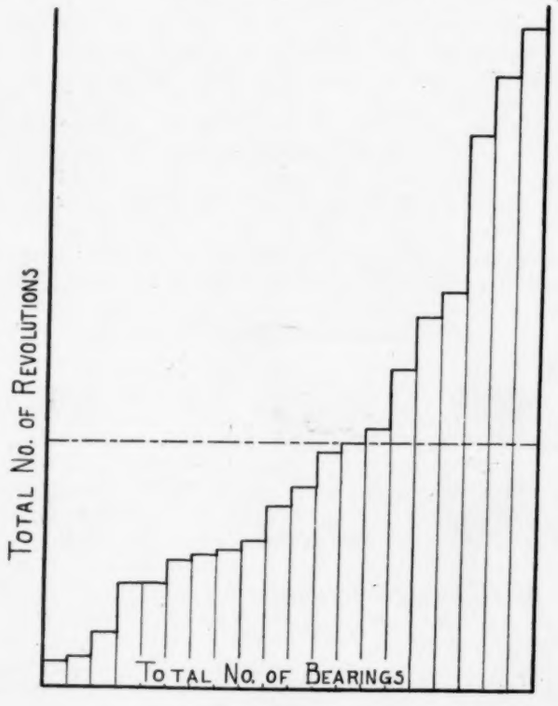


FIG. 5

hand is not subjected to any such limitation and its design and efficiency are benefitted accordingly.

Fig. 4 illustrates the application of roller bearings to a mill motor. These bearings are of the self-aligning type and the particular application is a changeover from plain bearings.

Development of Ball Bearings.

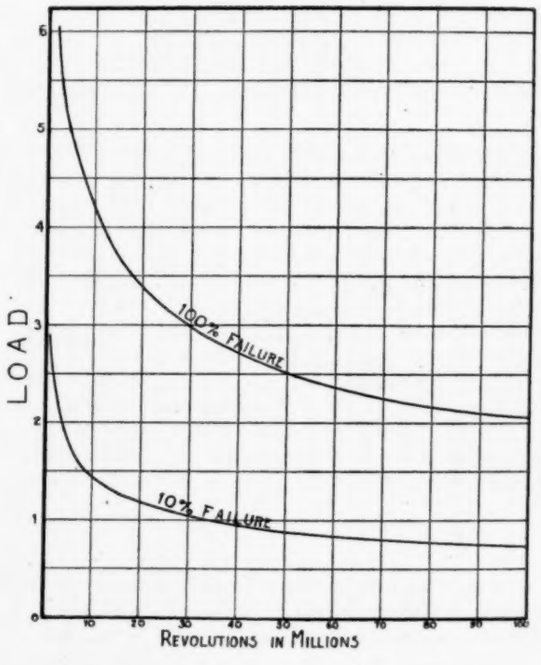
It has been said that great progress has been made in improving the design and quality of anti-friction bearings. At the present time, the quality of the material, the control of heat treatment and the precision of workmanship, consistently represent a higher standard than is attained in any other of the mechanical arts.

The necessity for this is apparent with even a superficial study of the mechanics of ball bearings. Contact pressures between ball and races will normally exceed 200,000 lbs. per square inch at the point of maximum stress. The distribution of load within the bearing, that is among the several balls, is absolutely dependent on the uniformity of ball diameter. Static analysis of this distribution fully justifies the actual manufacturing tolerances observed of plus or minus one-half of one ten-thousandth of an inch. Under such conditions it is seen that quality of material and precision of workmanship are vital to the successful performance of a bearing. In the face of active competition the utmost refinement must be maintained to insure this high standard of quality.

The performance of a bearing in service directly affects the interest of those concerned in maintenance. It will thus be of interest to mention the progress which has been made in determining the relation between load and life of ball bearings. Only a brief summary of this will be attempted.

In the first place it is necessary to understand in a general way the nature of bearing failures. There is,

FIG. 6



as in the case of plain bearings, a certain proportion due to abnormal conditions, such as the entrance of dirt and abrasive matter into the bearing. The effect of such on ball bearings is to produce wear as a result of the lapping action between balls and races. In time excessive clearance develops within the bearing and this becomes apparent through vibration. Such a failure evidences itself as a chronic symptom, giving adequate warning, and contrasts with the acute condition of heating and seizing in a plain bearing. Obviously it is difficult to foresee this class of failure and in the matter of design, nothing can be done beyond providing effective enclosures to anticipate the condition. There are other similar types of failure such as those caused by faulty lubricant or abuse of the bearing.

Contrasting with the erratic and indeterminate factors that limit the life of a ball bearing is the phenomenon of fatigue. This is a progressive form of failure, characteristic of the effects of repeated stress, and in a bearing first becomes evident as a flaking on the load surfaces of the race. When the high contact

stresses in the bearing elements are considered, together with the frequency with which they occur in a bearing operating at electric motor speeds, it is not surprising that fatigue plays an important part in limiting the life.

This fact has been generally known for some time, and the anti-friction bearing engineer has been accustomed to reckon with the situation by means of suitably proportioned load ratings taken with adequate factors of safety. Recently, however, the S K F Research Laboratory has made an exhaustive study of the fatigue phenomenon and compiled data from a very large number of tests. From this it is possible to formulate more definitely the load life characteristics of bearings.

From the operator's viewpoint particular interest attaches to the life dispersion of a given bearing. (See Fig. 5.) This refers to the variation in life of apparently identical bearings under identical load conditions. It has been found that some bearings fail earlier than the average, while others operate considerably beyond it. As an approximate relation 40 per cent of a given number of bearings show a life greater than the average. For example under certain conditions of load a number of bearings might show 10 per cent fatigue failure after 47,000,000 revolutions, while one bearing in a hundred would operate 1,040,000,000 revolutions. The average life of all the bearings would be about 500 million revolutions. This variation in life is typical of all standard types and sizes of ball bearings.

Another interesting fact in regard to fatigue (see Figs. 6 and 7), is that by reducing the load to one-half of a given value the life of the bearing will be increased 10 times.

These characteristics of fatigue failure should demonstrate the fallacy of attempting to draw general conclusions from isolated cases of failure. They also indicate the extensive service which the anti-friction

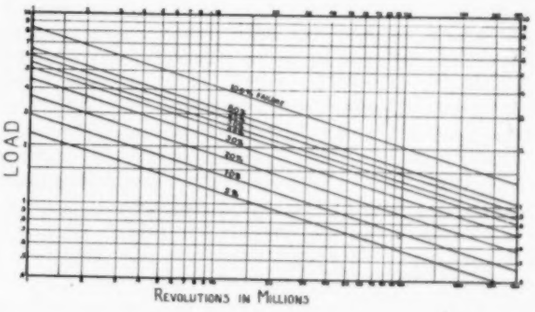


FIG. 7

bearing engineer can offer to both the motor designer and operator, in discovering and eliminating causes of bearing trouble.

In conclusion it may be said that anti-friction bearings afford a definite means of improving the operation of electric motors. The full realization of this possibility is largely dependent on co-ordinating the specialized knowledge of the bearing manufacturer with the extensive experience of the motor builder and operator.

Selection, Installation and Operation of Anti-Friction Bearings

By J. T. R. BELL*

THE selection of the proper type of anti-friction bearing, approached from a theoretical viewpoint, might lead to a highly technical debate or discussion covering, firstly, ball bearings vs. roller bearings, and, secondly, the various types of each; and as the "Rollway" bearing is not intended primarily to be a substitute for ball bearings, but rather a complement to them, I will, after a brief discussion of the various types, confine myself to the short roller or precision type, of which the "Rollway" is an example.

The ball bearing, while its variations are many, consists essentially of a number of balls or spheres running in a curved outer and inner race and capable of carrying a certain amount of thrust load in conjunction with its radial load; the sum of these two load components always being kept within the rated capacity of the bearing and vary in their relation to each other by increasing or decreasing the angle of contact, the plain radial bearing whose contact angle is 90 deg., or directly perpendicular to the axis of rotation being the type capable of taking the heaviest radial load. As we get away from this by inclining the angle of contact, the thrust capacity is increased and the radial capacity decreased proportionately. As originally made, the races of the ball bearing had a very much larger radius than the ball, resulting in a point or very small area of contact between the two, with consequent high unit stresses, making it impractical to use the same where shocks of any kind had to be contended with. It was later found that this condition could be reduced considerably by the use of raceways whose radii more nearly approached that of the ball, resulting in increased contact between the ball and race with consequently reduced unit stresses and increased capacity, and, of course, a corresponding increase in friction. I believe we can fairly credit American engineers for discovering that this increase in friction was more than offset by greater carrying capacity of the bearing, close contours between balls and races being now every-day practice.

It being impossible to go beyond a certain point with this method of increasing contact, it is here the roller bearing with its increased area of contact comes in. The ball bearing, of course, can be increased in size to effect the necessary contact area to support

*Rollway Bearing Company, Inc., Syracuse, N. Y.

the load and theoretically allow a sufficient safety factor for shocks. This, however, often calls for bearings whose diameters make them both extremely costly and unpractical from the designer's viewpoint.

Referring now to the roller bearing, fortunately or unfortunately, depending on whether you are a manufacturer or user of roller bearings, the same unanimity of opinion does not apply to them, there being two distinct types, namely, the taper roller and the straight or cylindrical roller, as well as a number of other types that might be called "bastard" types in that they are not strictly either roller or ball bearing, but rather compromises in design by which attempts have been made to incorporate the good features of both the roller and the ball bearing with the natural sequence that they are also subject to the shortcomings of both. I will, therefore, restrict myself to a short description of the two types most generally in use, namely, the taper and cylindrical.

The taper roller bearing consists of a conical roller running on an inner and outer race, these being also conical in shape and, due to the load angle, the bearing naturally cannot have the radial carrying capacity that the cylindrical roller has. In fact, as with the angular contact ball bearing, these bearings must be used in pairs (See Fig. 1) opposed to one another or some other method of keeping the bearing in place

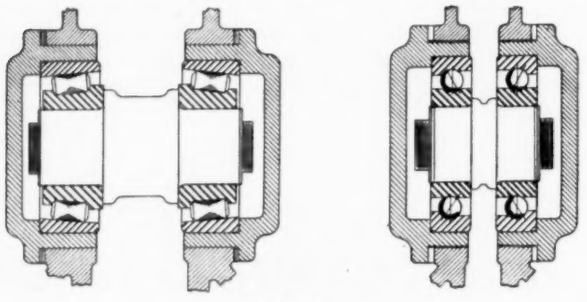


FIG. 1.

must be devised; otherwise, it would gradually work out from under the load, making this type unsuitable for certain classes of service, such as where there might be expansion due to heat or other causes or where accuracy of alignment is required, as the commercial taper bearing must be allowed a certain amount of end play to permit its free operation.

The cylindrical roller bearing carries its load perpendicular to its axis of rotation and cannot slip or slide from under same, and, having the greatest possible area of contact for any given length, it is therefore capable of carrying heavier radial loads than any of the foregoing types.

The cylindrical type of radial roller bearing or journal bearing, as it was generally known in this country, usually consisted of an inner and outer sleeve of any length between 2½ in. and 10 in., with a number of rollers of equal length assembled in some form of cage, and on these the load was carried. There being no commercial method of grinding these long rollers accurately, the bearing had necessarily to be either very crude or go to the other extreme and be very costly. Europeans found this out many years ago and resorted to short rollers whose diameter and length were practically equal; all rollers were originally ground on centers and, by reason of the difficulty in maintaining on the longer rollers accurate centers and the spring of the roller itself, accuracy, as already stated, was commercially impossible. On the other hand, this condition is not present in short rollers and the accuracy of same can be governed within the same limits as are balls in the ball bearing. In fact, today modern grinding methods have made it commercially possible to hold rollers to as

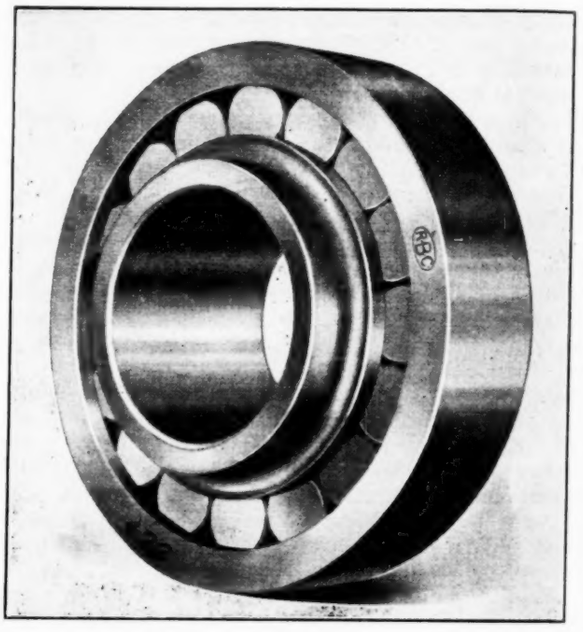


FIG. 2.

close and closer limits than the balls in the finer grades of ball bearings, the wrist pins in high grade automobiles being held to .00005 total variation in diameter, whereas, balls are usually allowed to vary as much as .0001 above and below normal. The accuracy obtainable in these short rollers more than compensates for the loss of capacity brought about by their decrease in length and has made the modern high speed precision type of roller bearing possible. (The term "precision" I use as implying accurate workmanship on rollers and races, thus insuring maximum contact between them, and interchangeability of complete units.) So much for types.

In the selection of a bearing, the nature of the work it is required to do should first be given careful consideration; that is, whether the load is a constant one with little or no chance for shocks, or a load in which shock is the major condition. Consideration should also be given to whether the load is principally radial, principally thrust, or a combination of the two, (in the majority of cases the latter condition will be found to be the more common one) as upon these facts would depend the type of bearing selected.

For constant steady loads where the shaft diameter does not run above 2 inches, the ball bearing will usually be found to give satisfactory service combined with a reasonable first cost. Shafts of 2 inches and upwards where loads become what is generally termed "severe" is the point where the difference of opinion becomes greatest and where, in our judgment, the roller bearing is preferable, the larger diameter of shafts usually implying the use of either chain or gear drives or double ply belts, the splice of which subjects the bearings to shock loads far in excess of those

shown by the ordinary method of calculating, and it is the frequency of such conditions that makes the use of the roller bearing with its greater area of contact safer practice.

Design limitations again often call for bearings of smaller outside diameter than is possible with ball bearings and here again the characteristics of the roller bearing are very much in its favor, as bearings of very much smaller diameter can be used.

In common with another manufacturer, we believe in a high degree of resiliency for the roller. This other manufacturer, however, obtains his resiliency by mechanical means, whereas the resilience in our roller is attained by the use of a special analysis of electric furnace alloy steel. I have possibly digressed a little by enlarging on this point; however, I want to stress the fact that nothing must be left undone in the manufacture of bearings to meet conditions in the steel mill industry as we know by experience they are. General-purpose bearings, while seemingly successful at the outset, have not the inherent stamina necessary for this service.

A bearing for this class of service should be simple both in design and method of mounting, also sufficiently rugged in its details so that it can be assembled and disassembled by the ordinary labor found in the electrical departments of steel mills without fear of injury, which includes damage in transit or accidental dropping of motor.

These features in the "Rollway" MDW (Fig. 2) type of bearing are accomplished by the wide inner race which is mounted a light driving fit, the use of lock nuts or retainers of any kind being optional. This light fit also allows of the bearing being removed readily by the aid of an ordinary puller. The wide inner race at the same time accomplishes another very essential function—that of preventing the bearing peaning into the shaft as has formerly often been the case with bearings having the narrower or standard width of inner race. The heavy section of this inner race also obviates the possibility of the operator cramping the bearing by driving same on too tight. The steel used in our bearings, being of the same analysis as that used in the standard ball bearing, is susceptible to this possibility by reason of its elasticity; case hardened bearings under such conditions usually break.

Maximum load capacity is necessary to withstand shocks and is attained by the use of the largest number of short square rollers guided on their ends by the channel section of the inner and outer races. No separator is used on our heavy duty types, which, as well as necessitating the leaving out of a number of rollers and decreasing the capacity, would introduce an element of weakness which sooner or later causes trouble, particularly in motors or other mechanisms which are subject to constant reversals.

To obviate the necessity of complicated expensive mountings or adjustments which invite trouble, the bearing must be self contained and also have the ability to locate shafts axially, and, if necessary, withstand severe surging end thrust. This, again, is taken care of in the "Rollway" design. The inner race on which the shocks come with greatest severity is, in the MDW type, of extremely strong section, the limiting feature to the amount of thrust that this race will stand being ability to maintain an oil filament on the end of the roll. No thrust bearings are required except under conditions entailing constant heavy thrust, such as bevel gears, etc.

The selection of this type of bearing, conforming as it does with the international standards for ball bearings, also allows for the widest possible number of options as, in case of fire or strike at the manufacturing plant, the user could always obtain some other bearing that would take care of his requirements, at least temporarily.

Installation.

The type of bearing now having been determined on, before installing same the loads the bearings will be required to stand should be carefully checked up to determine proper size to use.

There are several methods of arriving at bearing loads, that possibly most commonly used being the

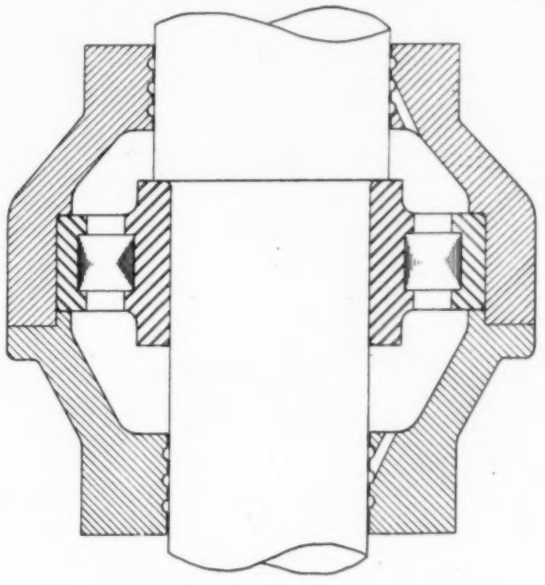


FIG. 3.

assumption of 200 pounds per inch of width for single and 300 pounds per inch of width for double belts, then by taking distance between bearing centers and distance to center of pulley, together with diameter of pulley, a set of dimensions is thus secured which with this belt pull can be used to compute bearing loads.

In most of the applications that we have made, shock load conditions have always been very much more severe than the computed loads. This, however, in the case of the roller bearing, does not require such careful consideration as it is designed principally with a view to meeting such condition, this feature having been developed to the highest possible degree.

Assuming that the shafts are of sufficient size to prevent excessive flexing or bending at the bearing seat, we are now ready to proceed with a description of the actual mounting of the bearing and it is at this point that our very extensive practical experience leads us to differ materially from most other manufacturers in that we believe any bearing suitable for severe service in mill type, railway or general-purpose heavy duty motors or other classes of machines should be of sufficiently rugged design that it can stand the brunt of every-day shop practice. Any

mechanism so delicate that its usefulness turns on the exclusion of a small amount of dirt or grit, too much or too little axial freedom, or the use of a certain brand of lubricant, has no place in a steel mill. Also, we do not favor either shrinking bearings on by expanding in hot oil, using arbor presses or resorting to other equally severe methods to get a proper seat, (our own experience shows that with a

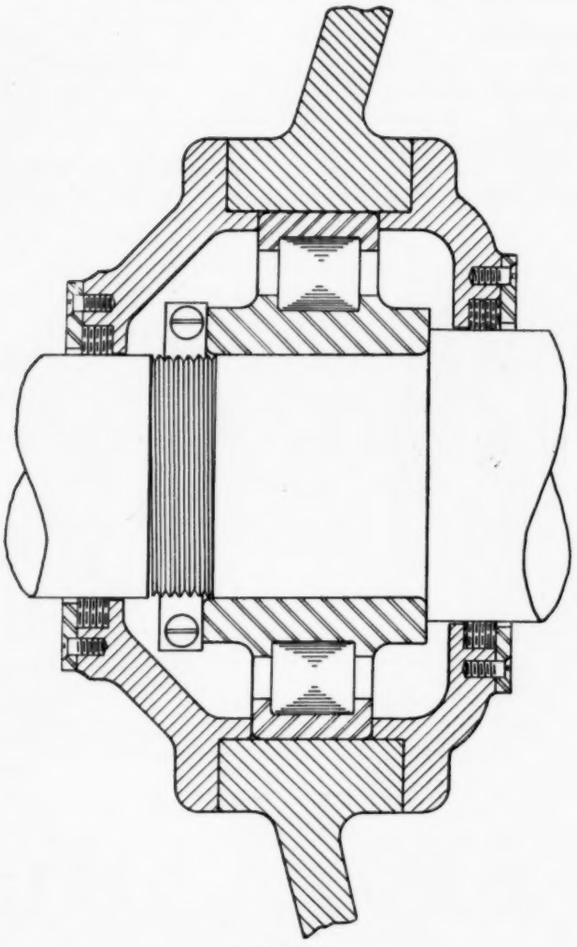


FIG. 4.

properly designed bearing, that is, a bearing whose inner race is properly proportioned for its load rating, this is not necessary), the possibility of overdoing this fit being very great, the result being a cramped bearing, and, granted that this does not happen often, heroic conditions have to be resorted to to remove the bearing.

The housings in which the bearings are placed should be of the simplest form possible. In mill type motors some form that allows for ease of removal should be adopted. In general-purpose motors the same principle applies, except that it must be accomplished in another way.

The following cuts illustrate a few of the types which we have found in practice to be very successful. In the mill types we have for a number of years used felt packings, which I grant are allowed to be "old-fashioned." However, they have demonstrated their ability to serve their purpose and serve it well.

A later form (Fig. 3), of which we now have a number in service, uses grooves instead of felt packing, and the results of this in service have been equally good with the former type.

In both of these designs simplicity has been the governing feature and we do not believe in labyrinth seals or complicated forms of slingers that could be put in in the reverse direction to what they are intended and, instead of deflecting the lubricant onto the bearings, might direct same away from them.

For induction motors we have adopted a different type of design (Fig. 4). In this we show a design for which we are indebted to our goods friends, the Westinghouse Company. This we have not used largely, as it is only useful in the case of oil lubrication, and as we do not favor oil for roller bearings up to speeds above 2,000 rpm., we have not found it often necessary to adopt this type of mounting. The outside chamber in this housing, which you will notice, is intended for an expansion or settling chamber for the oil. In the case of very high speeds, if the lubricant (which would likely be oil) is allowed to become too high on the bearings, there is a heating and emul-

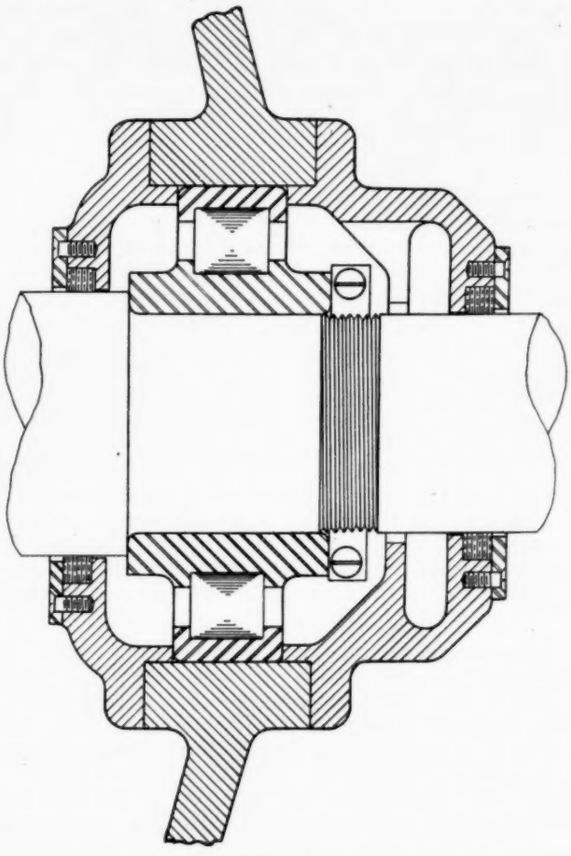


FIG. 5.

sifying effect. The use of this chamber largely overcomes this condition and also prevents leakage.

The other mounting shows the type (Fig. 5) which we favor for general-purpose motors. This mounting allows for the armature being removed from either end without disturbing the other end, the caps in both types being retained by bolts fitting into holes drilled through the end bell of the motor and allowing, as already stated, for removal readily in either direction.

We are now at the stage where the bearing has been selected, as also the type of housing and the whole assembly ready for service. The question of lubrication now comes up.

As anti-friction bearings have been adopted by steel mills, first, because by their use it is possible to overcome one of the principal difficulties experienced by the operator, namely, slinging or throwing of oil which results in electrical failure of the motor; secondly, lack of any oil at all, which results in mechanical failures; and thirdly, the question of upkeep. Speaking for roller bearings, as stated earlier, we prefer the use of a neutral grease about the consistency of vasoline with, of course, a higher viscosity. For speeds above 2,000 rpm. we have found oil to be most suitable. We do not lay down the use of grease as a hard-and-fast rule for the lower speeds, as we have a number of installations in operation which are successfully using heavy oil, the operation of which we will describe later. High pressure grease systems are not advisable on motors, as such pressure can easily be exerted as wil force the grease through the seals; a low pressure grease gun, however, is favored in that it tends to keep the lubricant clean.

Operation.

Before getting down to a description of some actual operating conditions on mill motors, I will give a short history of our bearing.

The Railway Roller Bearing Company (of which the Rollway Bearing Company is the successor) started the manufacture of journal roller bearings for use in the journals of railroad equipment about 15 years ago. The result in the earlier types being rather disastrous, an analysis of the situation brought about the adoption of two shorter roller bearings in place of one long one; this proved a step in the right direction (some of the failures that then occurred, in my opinion, were due to the generally limited knowledge of the art of bearing design and manufacture compared to our knowledge of the same subject today, the principal of these limitations being the types of steel that were at hand to manufacture the bearing from and limited knowledge of heat treatment.)

Our experience with these short rollers was so encouraging that they were still further shortened with increasingly satisfactory results and without, I believe, any consciousness on the part of the designers in our company (as they were principally concerned with their own product and did not give much attention to what anyone else was doing) that they were working along parallel lines with those that were proving themselves successful in Europe.

About this time a number was installed in electric railway motors, some of which were in operation for many years and a few of which I believe are in operation still. This development brought about others that in turn lead up to the first industrial installation in an electric motor made 13 years ago, the motor being a General Electric 30-hp. Type K motor in use by the Solvay Process Company, this motor being geared to large drums used in the drying of soda ash. Pulverized lime, ammonia fumes and high temperatures are present in this installation and lubrication previously was impossible .

This motor was rewound about a year ago and the bearings inspected and put back in service and in a few weeks they will have completed their thirteenth year. The motors in question run about 900 rpm.

The success of this installation led to them gradually equipping other motors, till now they have approximately 25 in service, the majority of these having given six years or more of continuous service, with the exception of a few installations made recently.

Following this, installations were made in General Electric Type KT Induction—2 hp. to 150 hp. and

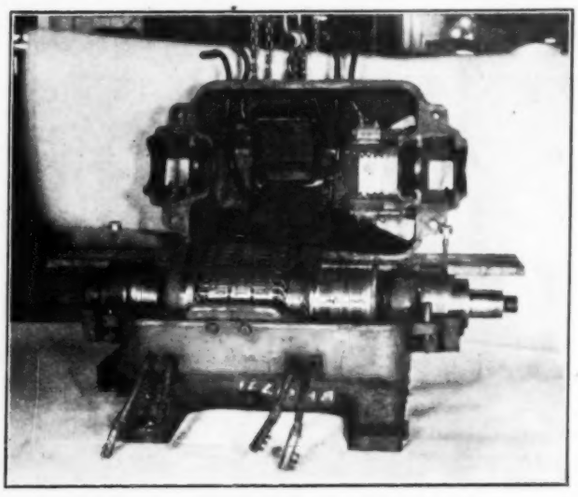


FIG. 6.

Railway 88, 200 and 300 series; Westinghouse Type K crane motors from 3 to 30 hp.; Railway motors Types A and J, and Type MS induction motors from 1 to 25 hp.; Allis Chalmers induction motors 20 hp.; Burke induction motors from 5 to 40 hp., and Reliance motors; also Crocker-Wheeler "W" types mill motors from 12 hp. to 150 hp. and induction motors from 2 to 30 hp.; and later General Electric and Westinghouse Company mill types.

As the gradual evolution in design increased the bearings' ability to take care of much heavier loads than had hitherto been considered possible with anti-friction bearings and thus earned a growing confidence on the part of the user, this, in turn, led to its installation in many places in which no type of bearing—either anti-friction or sleeve—had ever given satisfactory service previously. One of these applications, namely, that of the G. E. MD 108 100-hp. motor used

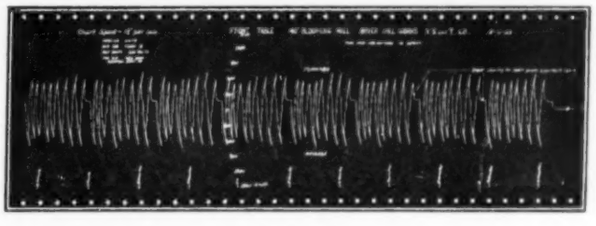


FIG. 7.

to drive the reversing front tables in a 40-in. blooming mill—a service the severity of which most of you engineers are, no doubt, familiar with.

Fig. 2 shows a photograph of the bearing, namely, the "Rollway" MDW 324, and a cross-section of same, a description of which has already been given.

Fig. 6 shows the motor opened up, giving a very good idea of method of installing bearings.

The 100-hp. motors driving screwdowns on this mill are also equipped with "Rollway" MDW bearings.

This particular installation completed its first year's continuous service without delay of any kind December 28, 1923. In the course of that time the bearing housing, we understand, has never been opened, even for inspection.

Fig. 7 shows a current chart taken from this same motor and, should there be any among you who are not familiar with steel mill service, the severity of

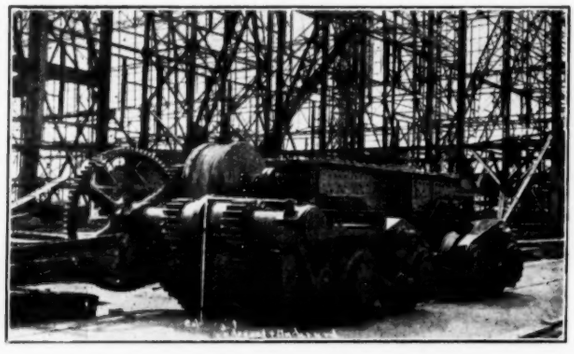


FIG. 8.

same is graphically depicted in this chart and, I believe, carries out our company's claim that applications of this kind are no longer experimental; and that this type of anti-friction bearing is a thoroughly tested and proven product, capable of taking care of loads, no matter how great the severity, when properly applied. In fact, anti-friction bearings are even more necessary for heavy power drives, railway and mill service and all bearings carrying severe loads, than they are for lighter duty where ball and roller bearings are now considered essential, the trouble being that suitable bearings were not formerly obtainable.

As a matter of general interest, I have added the following illustration:

Fig. 8, which depicts one of the trucks that carry the doors of the Lakewood hangar in which is housed the U. S. S. Shenandoah, each door of this hangar weighing 3,500 tons or 7,000,000 pounds. The doors are 135 feet high, 165 feet wide and 15 feet thick at base.

I could go on indefinitely describing motors equipped with this bearing on cranes, mill tables, centrifugal pumps, motor-driven screwdowns, rod mills and other types of ore crushing and pulverizing machinery, together with many installations on heavy calendering rolls in which great heat and moisture are present, as well as enormous radial loads. I trust, however, that the installation described and the others depicted have given you a very good idea of the capabilities of anti-friction bearings when intelligently selected for the work which they are intended to do.

Before closing, I would like to state that I am indebted to Mr. Lawrence Hess of Youngstown Sheet & Tube Company, Mr. C. W. Howgate, General Electric Company; Messrs. Hipple and Pruger, Westinghouse Company; Mr. Dobelblower of the Crocker-Wheeler Company, and Mr. McCutcheon of the Reliance Company, for suggestions from time to time which culminated in the mountings illustrated; and for the pictures and information covering the operating conditions Mr. R. E. Lewis of the Brier Hill Division, Youngstown Sheet & Tube Company, as it was through his courtesy we were able to present these very interesting pictures.

Mounting and Maintenance of Roller Bearings in Electric Motors

By U. B. WACHTLER*

SUMMARY

Mounting.

Requirements of the Bearing:

Size of Bearing.
Alignment.
Lubrication and Cleaning.
Exclusion of Grit-Retainment of Lubricant.
Location of Shaft Endwise.
Fit of Bearing on Shaft and in Bore.

Operating Demands on Motor Affect Design of Bearing Mounting:

Mill Type Motors.
General Purpose Motors.
Standardization of Bearing Sizes.
Crane and Hoist Motors.

Maintenance.

Length of Life of Roller Bearings.
Lubrication.
Flushing Out Housing.

The factors which most directly influence the mounting of roller bearings in electric motors are (1) the requirements of the bearing itself for successful operation and long service, and (2) the demands which will be made on the motor as regards the nature of the work it must do, and the required ease of inspection, repair and replacement and of general accessibility and convenience.

Requirements of the Bearing.

There are several fundamental conditions which must be observed in order that roller bearings may operate successfully in electric motors over a long period of time. These will be considered separately in some detail, since a clear understanding of these

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requirements will simplify the problem of proper bearing selection and mounting.

Size of Bearing.

The first point to be observed is that the bearing be of the proper size. The process of choosing a bearing for a given motor generally has for its first step the placing of some value on the load which will be imposed on the bearing, and a similar evaluation of the ability of the bearing to carry this load. Both of these are processes not entirely removed from the field of speculation, as anyone who has made such computations must have observed.

There are several methods of approximating the loads imposed on the bearings of electric motors. In the order of their reliability, as we regard it, these are as follows:

(1) The capacities of the plain bearings on motors of existing design. Years of experience have shown the motor manufacturers how large a plain bearing is required in each size and type of motor, and what pressures per square inch of projected bearing area can safely be imposed. By using average values of bearing diameter and length, and a constant or gradually increasing unit pressure as the bearing diameter increases (100 to 1250 lbs. per square inch is a reasonable range), it is possible to arrive at the total bearing capacity which must be provided in any given motor, whether plain bearings or anti-friction bearings are used. This method of evaluating bearing loads, although strictly empirical, has the advantage of using for its basis the results of years of actual experience in motor design; consequently, the figures obtained by its use can be credited with a high degree of reliability.

(2) The strength of existing armature shafts. For ultimate economy of design there should be a balance between the bearing capacity and the ability of the shaft to utilize that bearing capacity without exceeding certain limits of stress and deflection. Inasmuch as the Hyatt bearing as applied to electric motors permits the use of shafts whose dimensions and proportions vary but slightly from existing plain bearing shafts, this relationship or balance affords a valuable check on the bearing load. For combined torsion and bending, a conservative figure for the allowable stress in the shaft is 8,000 lbs. per square inch. The conservatism of this figure makes unnecessary any further allowances for shocks and overloads.

(3) Total belt pull, usually assuming a combined tight and slack side tension of 200 pounds per inch of width for single belt and 300 to 350 pounds per inch for double belt. By taking average distances between bearing centers and average distances from bearing to pulley centers for most of the different makes of general purpose motors, a set of dimensions is secured which can be used in conjunction with this belt pull to compute the bearing loads. This method takes no account of the power delivered by the motor, but considers only the tightest belt that could reasonably be expected.

(4) The power delivered by the motor to the pinion or belt. This involves the motor rating, speed and diameter of pinion or pulley and results in a figure that must be corrected for the tooth form in the case of the pinion and converted from effective belt pull to total belt pull in the case of the pulley. To all of the above figures it is, of course, necessary to add vectorially the weight of the armature and any allowances for unbalance of rotor, shock load, etc.

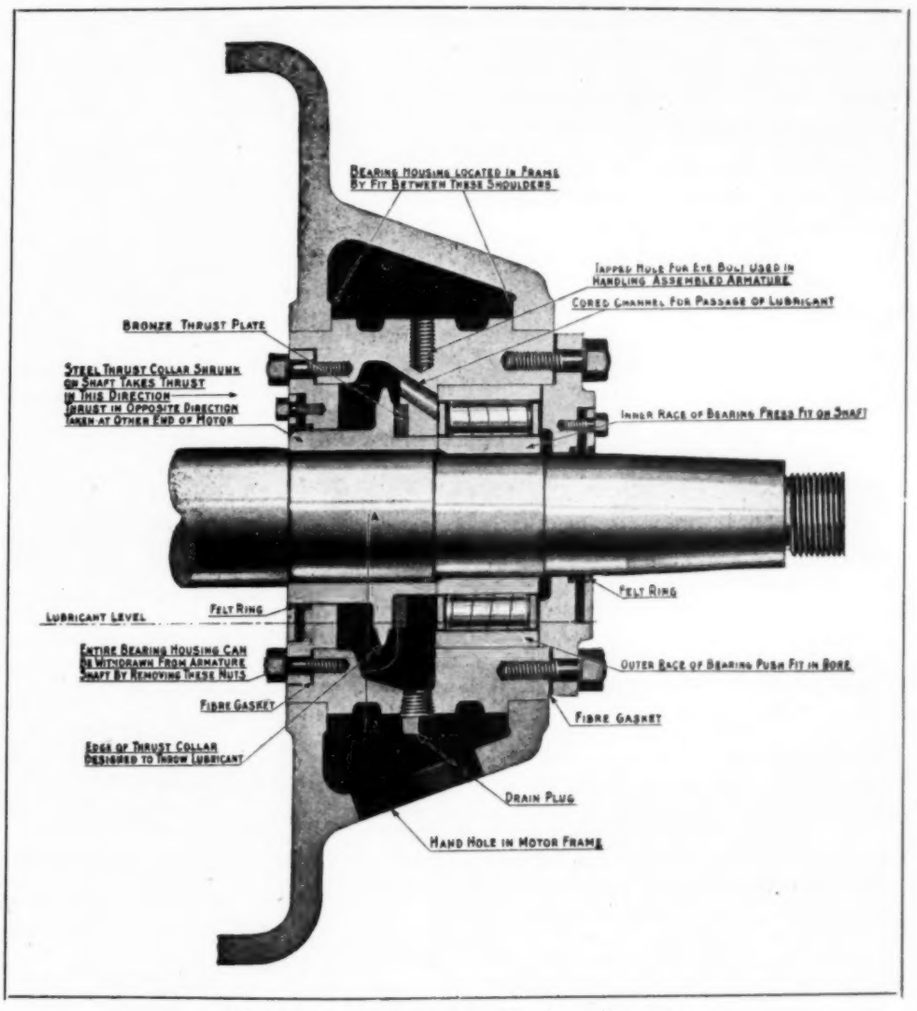
Bearing Housing Located in Frame by Fit Between These Shoulders
Tapped Hole for Eye Bolt Used in Handling Assembled Armature
Cored Channel for Passage of Lubricant
Bronze Thrust Plate
Steel Thrust Collar Shrunk on Shaft Takes Thrust in This Direction
Thrust in Opposite Direction Taken at Other End of Motor
Inner Race of Bearing Press Fit on Shaft
Felt Ring
Lubricant Level
Felt Ring
Entire Bearing Housing Can Be Withdrawn From Armature Shaft By Removing These Nuts
Outer Race of Bearing Push Fit in Bore
Fibre Gasket
Fibre Gasket
Edge of Thrust Collar Designed to Throw Lubricant
Drain Plug
Hand Hole in Motor Frame

FIG. 1—*Typical Hyatt mounting for mill motor.*

Considerable experience with bearings in electric motors has convinced us that there are so many other factors beyond those already considered which influence the bearing load, and their possible effect on the magnitude of the load is so great, that any values computed according to methods similar to the above are

of relative or comparative value only and cannot be used as an absolute measure of bearing loads.

For instance, a poorly designed or applied belt lacing or fastener will set up a shock load which places heavier demands on the bearings than the entire computed belt pull. A new pinion and gear on accurately maintained center distances will develop only a fraction of the bearing load resulting from worn gears mounted on loosely fitting shafts.

The heaviest demands made on the bearings of an electric motor are in the nature of shock loads or suddenly applied loads which are far in excess of the normal operating load on the bearing. Consequently, the overload and shock load capacity of the bearing becomes an important factor in determining the size bearing required. In the Hyatt Bearing, due to the nature of the roller, the stresses are far below the elastic limit of the material, thereby leaving a large margin for carrying the shock loads. In types of bearings where the stresses are far higher, the insurance against shock loads and abuse must be secured by increasing the size of the bearing.

Due to the fact that the same size frame is used with two or three different sets of windings to produce motors having different ratings according to their operating speeds, it becomes necessary to provide a bearing which will satisfy the requirements of all of them in order that frames and bearings can be used indiscriminately. Usually the bearing size will be determined by considering the winding with highest speed and highest power rating, since the mechanical stresses and bearing loads are approximately the same for all three ratings whereas the bearing capacity drops off as the speed increases.

After consideration of the various questions above outlined has yielded some certain size of bearing, the general motor construction must be examined to see if this size is acceptable from all other angles. The strength of the shaft in bending might very possibly necessitate a larger diameter at the bearing seat and consequently a larger bearing than load considerations alone would require. The standard shaft extension diameter or the required diameter of the tapered pinion fit might be the limiting factor. The avoidance of sudden changes in shaft diameter to eliminate localized stresses might dictate the various diameters, beginning with the standard extension diameter and working toward the center, and thereby prevent the use of a bearing which required too sudden an increase in shaft diameter. Very frequently the weakest spot is the inability of the shaft to carry the heavy bearing load on the small surface under the inner race of narrow types of bearings. In such cases the metal of the shaft is peened down by the inner race, with the result that the race becomes loose on the shaft, a condition that results in rapid destruction of the whole bearing mounting.

In general, it can be said that for best adaptability the roller bearing should fit on a shaft diameter only slightly larger than the shaft extension diameter, and should derive sufficient capacity from its length rather than from increasing the diameter of its rolling elements. Space lengthwise of the shaft is not closely restricted except in very special motors such as mine locomotive motors, whereas a larger outside diameter of bearing is objectionable in almost every case because of closing up ventilating ports, increasing the weight of motor, etc.

In laying out a series of Hyatt Bearings for electric motor service, the best average values of bearing loads were secured by taking a weighted mean of results obtained by all four methods of computation described above and adding to this such values for shock, vibration, overload and abuse as our experience with electric motor installations over a period of years has shown to be necessary. Then for each shaft size a bearing was designed of such length that its capacity was ample for the respective motor service.

Proper Alignment.

Equally important with the size of the bearing is the provision for proper alignment. To satisfy this requirement necessitates a motor design of such character that the bearings are properly aligned to start with, and a sufficiently rigid construction of the motor frame to insure the permanence of the original alignment. In order to be commercially practicable the necessary degree of accuracy of workmanship must be obtainable by the use of modern production methods, without necessitating any unusual processes in manufacture.

In the Hill Type motor, examples of which are shown in Figs 1 and 4, the frame is split, and the bearing housings are clamped between the two halves of the frame. Each housing has a machined cylindrical seat at each end, and these rest in cylindrical saddles in the motor frame. In this way the housing is held concentric with the motor frame, and any tendency on the part of the housing to tip over is prevented by making it of sufficient axial length to insure a good stable construction. The bearing housings are prevented from moving endwise out of their normal position by means of flanges on their outer surface which fit between shoulders in the motor frames. By boring out both ends of the motor frame at one setting, accuracy is assured within the limits of the boring operation which is usually sufficiently close for the incorporation of roller bearings.

In general purpose motors, as shown in Fig. 2, the alignment of the bearings is secured in a different way. The ends of the frame are planed off, and counter bored. To each side of the frame is bolted the machined end bell or shield, which carries the bearing housing. The shoulder which is machined on the end shield fits into the counter bore of the frame, thereby preventing the bearing from moving out of its correct position. It is, of course, necessary to bore the end shields perpendicular to the plane of the face of the shield with a reasonable degree of accuracy, and also to locate this bore concentric with the shoulder which registers into the frame.

Motors which are designed primarily for crane and hoist service have some of the characteristics of both the mill motors and general purpose motors in respect to the support and alignment of the bearings. The frame is split, but as a rule the bearings are not clamped between the halves of the frames, but are bolted against the ends of the frame after the two halves have been fastened together. The elements of the design which insure the alignment are practically identical in principle with those which are characteristic of the general purpose motor.

On the whole, it can safely be stated that the incorporation of roller bearings into electric motors does not require any radical changes in design or accuracy of manufacture from the present plain bearing

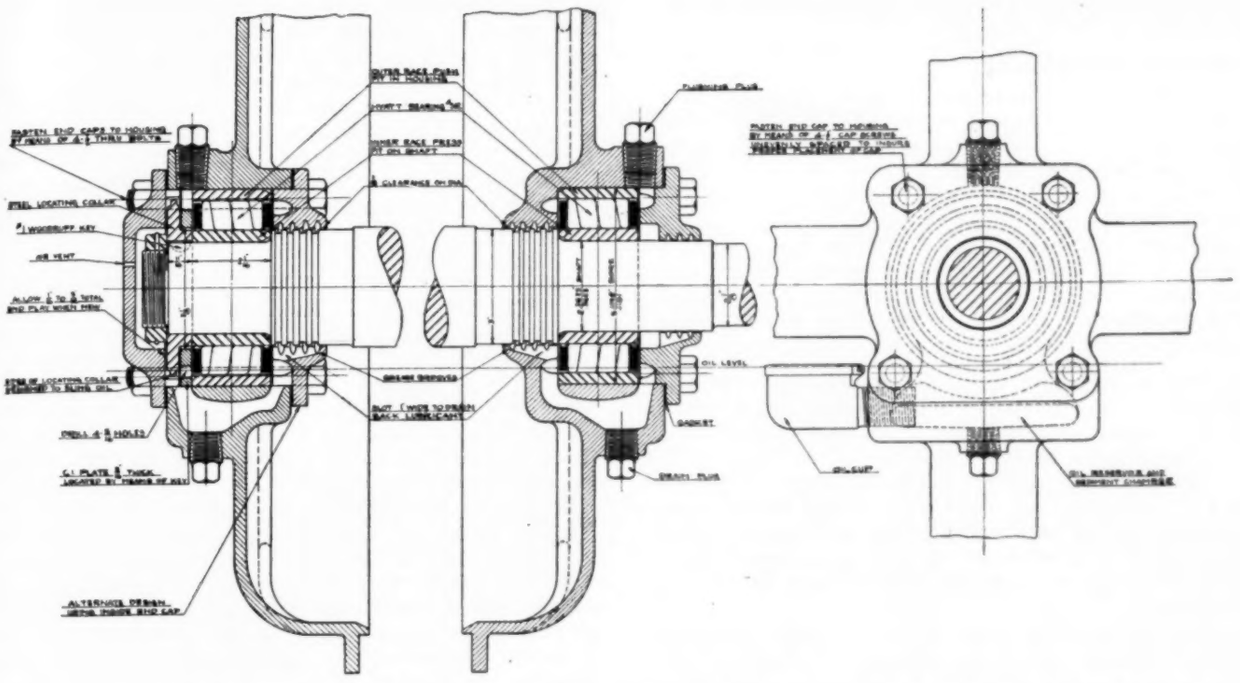

FIG. 2—*General purpose motor—motors according to similar designs have operated more than eight years without repairs or replacement of bearings or any attention other than lubrication.*

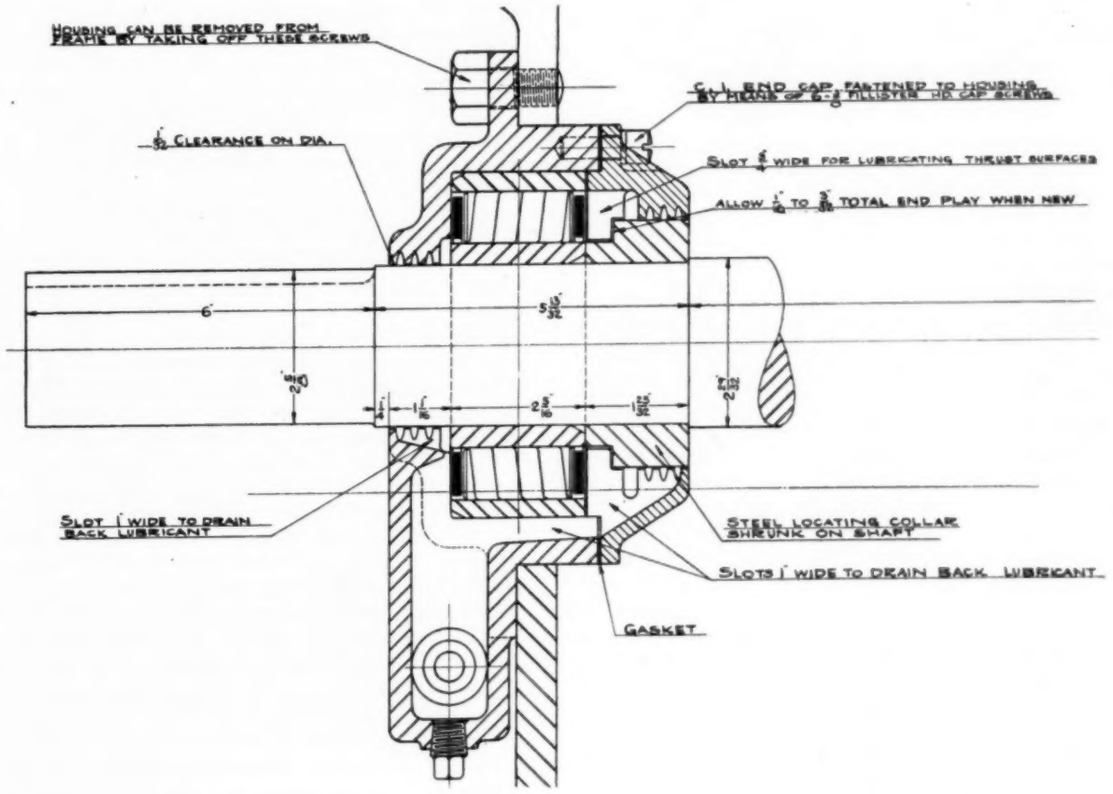
HOUSING CAN BE REMOVED FROM FRAME BY TAKING OFF THESE SCREWS
C.I. END CAP FASTENED TO HOUSING BY MEANS OF 6-⅝ FILLISTER HD CAP SCREWS
½ CLEARANCE ON DIA.
SLOT ⅛ WIDE FOR LUBRICATING THRUST SURFACES
ALLOW 1/16 TO 3/16 TOTAL END PLAY WHEN NEW
6
5 15/32
SLOT ¼ WIDE TO DRAIN BACK LUBRICANT
STEEL LOCATING COLLAR SHRUNK ON SHAFT
SLOTS ¼ WIDE TO DRAIN BACK LUBRICANT
GASKET

FIG. 3—*Bearing housing for crane motor—this housing fits into same space occupied by plain bearing housing and requires only slight machining of the shaft*

construction, insofar as features of alignment are concerned.

Lubrication and Cleaning.

The greatest advantages of roller bearings over plain bearings in electric motors lie in their lubrication requirements, not only as they affect the operation of the bearings, but also in eliminating the damage which is done to the motor insulation. Records indicate that the cost of the lubricant and the expense of applying it to a roller bearing motor is less than a quarter of the figure for a plain bearing motor, because of the fact that the roller bearings require lubrication and attention only two or three times a year instead of once a week or even more frequently.

However, the real advantage is of much greater importance than this. There is hardly any question that the largest item of cost in the upkeep of motors in steel plants, is that of repairing or replacing worn bearings, scored shafts and burned-out armatures and field windings. A large part of this expense can be traced directly to the absence of satisfactory lubrication where it is required, and the presence of lubricating oil or grease in places where it can do nothing but harm.

Properly selected and mounted roller bearings eliminate almost all of this trouble and expense. To begin with, the rate of wear of a properly functioning roller bearing is so slow as to be practically negligible for electric motor service. It uses oil very slowly and it does not "wear out" the lubricating qualities of the oil nearly as quickly as plain bearings. The slow rate of wear of the roller bearings makes it possible to design housings whose end plates fit very close to the shaft, since no allowance need be made for wear in the bearings. As a result, much tighter housings can be installed with roller bearings than have been possible in the past with plain bearings, and grit is kept out of the bearing and the lubricant prevented from escaping. An examination into the causes of motor trouble in various steel plant electrical shops makes possible the statement that the repairs due to oil soaked insulation are among the most frequent as well as the most costly. The use of better housings and better enclosures, made possible by the incorporation of roller bearings, will practically eliminate this item of maintenance expense.

In addition, burn-outs, caused by the rotor coming in contact with the pole pieces due to worn bearings are made almost impossible. The microscopic wear of the roller bearings holds the rotor central in the frame and maintains a uniform air gap indefinitely.

The question is frequently asked whether oil lubrication or grease lubrication is preferable for roller bearings. As far as the successful operation of the bearing itself is concerned, both oil and grease have proven themselves perfectly satisfactory, although it is self-evident that the lubricant should be of lighter consistency for the higher speeds of operation, in order that the friction and heat generated by churning up a heavy grease may be avoided. If the grease is too heavy, or if a drop in temperature causes a decided increase in stiffness, there is danger of the bearing forcing the grease out of its path and making a channel through it. In such event there is practically no real lubrication of the bearing.

There are, however, certain considerations of motor operation which have made it appear to us that oil is more desirable than grease. Over-lubrication is probably the largest single cause of motor troubles, and over-lubrication can be more readily avoided with oil than with grease, due to the fact that oil can be poured and will establish its own level, where as at ordinary temperatures this is not possible with grease. Figs. 2 and 4 show the manner in which this fact is used to advantage in designing a bearing housing for a general purpose motor and a mill type motor. It will be noted that in each case the oil is introduced through an opening placed exactly at the desired lubricant level, and consequently it is impossible to introduce too much lubricant. In this way, flooding the bearing is prevented, together with all of its disastrous consequences.

It will be noted that in both of the designs shown there is a considerable reservoir of oil out of reach of the bearings, and the question might be raised as to what object there was in providing so large a supply of apparently inaccessible lubricant. The point is that due to the action of the rollers, the oil is continually circulated, and although at any one time only a small portion of the total actually comes in contact with the rollers, nevertheless all of it is used, and the rate at which it deteriorates, due to the presence of grit or to evaporation, is much slower where the quantity of lubricant is large, than if it were much smaller.

In the design shown for the general purpose motor, the oil is introduced through the oil cup, whose connection with the oil reservoir is well below the oil level. This fact makes it impossible for surface froth or bubbles to float out into the cup and possibly leak out.

The same question of over-lubrication makes it appear that any system for introducing the lubricant into the bearing housing under pressure, such as with a grease gun or similar device, should be avoided, since it is impossible to determine when the right amount of lubricant has been forced into the housing. A further disadvantage is the fact that if oil or grease is introduced in excess of the capacity of the housing, the full pressure of the grease gun is available for forcing the lubricant out past the seals, and into the motor windings.

Anyone familiar with the work of steel mill electrical engineers knows that there is a strong sentiment among them in favor of grease lubrication for antifriction bearings. This is largely a result of their unfortunate experience with some of their oil-lubricated plain bearings, from which, because of bearing wear or other causes, the oil escaped into the windings. It seems only natural for them to prefer the use of a lubricant which "stays put" and which is apparently easier to keep out of the windings, but we are convinced that the ease of lubricating motors with oil, the elimination of over-lubrication by proper use of overflow ports, the ease of keeping roller bearing housings oil tight, and the greater uniformity of oils over greases will be demonstrated to their satisfaction by the operation of one or two roller bearing motors on an oil or liquid grease basis.

The requirements of Hyatt Bearings as regards cleaning are very simple. Every year or so the bearing housings should be flushed out with kerosene or a similar solvent, in order to remove the oil which has gummed, and any grit or worn particles that have accumulated. This is provided for in the housings shown by a large plug in the top and in the bottom.

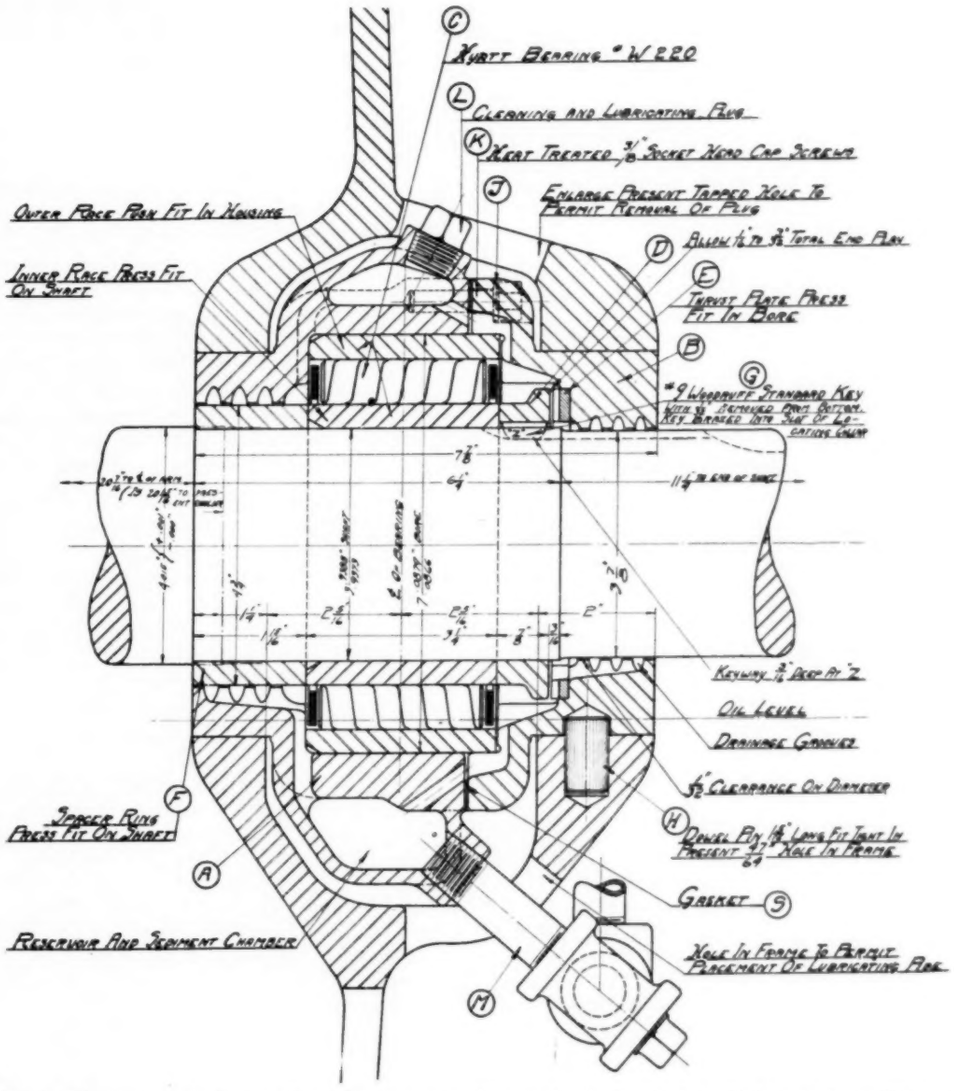

FIG. 4—*Changeover design — Hyatt bearing mounting designed to fit into plain bearing mill motor frame with no machining of frame and only slight changes to existing shaft.*

Exclusion of Grit—Retainment of Lubricant.

In order that the fullest advantages of roller bearings in regard to lubrication may be realized, the bearing housing must be tightly sealed to prevent the passage of lubricant out of the bearing and the entrance of dirt in the housing. The design of a suitable enclosure usually embodies a very small clearance between the moving shaft and a stationary part of the housing, or a fibrous or plastic packing of some kind, or a combination of both. Since one of the principal requirements of such an enclosure is extreme simplicity, the demands of practicality place a very early limit on all "devices" which seek a solution by multiplicity of parts.

Some of the most generally used seals or enclosures are shown in the various layouts. One of the simplest consists merely of a wide end plate having a clearance of about one sixty-fourth of an inch to one thirty-second of an inch on the diameter over the shaft size. Two or three oil grooves are machined into the inner surface of this end plate, and the walls between the grooves are cut away at the bottom by a narrow slot for the purpose of draining back whatever oil has found its way into the grooves. Sometimes a corresponding set of grooves is cut into the shaft directly underneath the grooves in the end plate, to assist in preventing the passage of oil.

An arrangement of this kind, with grooves only in the end plates, has given very good results in an application where it might be supposed that a far more elaborate seal would be needed. On four General Electric motors, 15 hp., rating, at 750 r.p.m., operating cylindrical coke screens at the Republic Iron and Steel Company, at Youngstown, O., Hyatt Bearings, protected by such a seal, have operated for several years, the bearings in one motor having been placed in service early in 1919. In installations where a simple seal of this kind will prove satisfactory, it is very desirable, since there are no packings to be inspected or replaced, and no attention whatever is required.

Where dust conditions are sufficiently unfavorable to make it appear that a more effective method of exclusion is required, the design shown in Fig. 7 can be used to advantage. This combines a very close clear-

ance at the inside of the seal, with a felt ring at the outside. This felt ring is cut so that its inside diameter is approximately one-quarter of an inch smaller than the shaft. This will cause the edge of the felt to bulge or spread, which action is provided for by leaving about one-eighth of an inch clearance between the shaft and those portions of the end plate immediately adjacent to the felt. A felt ring which is given a little space in which to expand will not become hard and cut the shaft as readily as it would if the felt were closely confined and tightly packed. It has been found helpful to boil the felts in melted heavy grease, which saturates them thoroughly and provides sufficient lubrication to prevent scoring the shaft or burning the felt. A feature which will assist in keeping this seal in good working order, is the fact that the felt can be inspected and replaced, if necessary, without disturbing the shaft, pinion or bearing, by simply removing the retaining ring which holds the felt in place.

One of the most common forms of seal combines a close clearance with a single large groove, which is generally packed with a felt ring. When new, this seal is reasonably effective, but it places the felt at a disadvantage, inasmuch as there is no room for it to exert the elasticity which makes it a suitable material for this purpose. In addition, there is no way of inspecting or replacing the felt without removing the end plate from the shaft. A very effective seal can be obtained by omitting the felt, and connecting a grease cup to this groove. The cup is given a turn at intervals of a week or so in order to keep the groove filled with grease.

The danger of this type of seal is that the excess grease forced out by too frequent turning of the grease cup will get into the windings and thereby defeat one of the main purposes of roller bearings. There is also a possibility that the workman who lubricates the motors will neglect the lubrication of the bearing itself, thinking that the grease cup which feeds the seal is all that requires replenishment.

Labyrinth seals are frequently used, and are quite effective in excluding the coarser forms of grit. The objections to this type of enclosure are the cost of machining accurately enough for effective performance, and the great danger of incorrect assembly after a motor has been repaired, due to the complex nature of the packing. Frequently, the design is such as to cause the packing to act as a pump, if there is any marked tendency for the shaft to oscillate endwise. This tendency would, of course, defeat the entire purpose of the seal.

Location of the Shaft Endwise.

The straight cylindrical construction of all parts of the Hyatt Bearing permits a reasonable lengthwise variation in the manufacture of shaft, frame and end shields without necessitating any adjustment whatever to the bearings. Both inner and outer races are wider than the effective roller lengths by amounts ranging from ¼ in. for the smaller sizes to ¾ in. for the larger ones. This fact permits variations from plus or minus ⅛ in. to plus or minus ⅜ in. in the machining operation which locates the bearing centers. These permissible variations are far in excess of ordinary production tolerances of the various parts which are affected.

The mountings of Hyatt Bearings have been designed to take full advantage of this fact. In designing a mill motor bearing layout, as shown in Figs. 1, 4, and 7, a locating collar is placed on the shaft near each end, its position being determined by a shoulder on the shaft. This collar carries a flange which operates against a lubricated surface and fixes a limit to the end motion of the shaft in that direction. Motion of the shaft toward the right is limited by the contact of the locating surfaces in the right hand housing; motion toward the left by the surfaces in the left hand housing. The distance between the thrust flanges on the shaft is made sufficiently smaller than the corresponding distance between thrust surfaces on the frame, to allow for variations in machine work, for any desired amount of end float of the armature, and for unequal expansion of the shaft and frame, due to temperature changes. The collar is generally hardened steel, and the surfaces against which it operates are either cast iron or bronze. All surfaces that come in contact and between which there is relative motion, are placed where lubrication is easy and positive.

In the general purpose motor, shown in Fig. 2, the motion of the shaft in both directions is taken care of by a single hardened steel collar secured by a nut at one end of the shaft, generally the front end. This collar operates between two machined cast iron surfaces. The distance between these surfaces, minus the thickness of the collar is the total end motion of the shaft; this can easily be adjusted to any desired amount by the addition or removal of shims under the housing end cap. Complete coincidence of the magnetic centers of the rotor and the field can be secured by shifting the position of the collar lengthwise along the shaft.

In both of the designs shown, the area of the surfaces which carry thrust or locating loads is ample not only for service conditions, but also to withstand the shock and abuse to which the motor is subject in transportation and in the course of mounting. Carelessness on the part of workmen who change pinions or pulleys on the shaft, will not damage the Hyatt Bearing, since with either of the designs shown it would be possible to drive on the pinion or pulley with a hammer and still carry all of this hammer blow on the steel collar. The liability of damage to the motor in handling with a crane, or by striking the exposed ends of the shaft, is practically zero.

Very heavy axial loads, many times as great as the computed radial loads on the bearings, are set up by the unequal expansion of the shaft and frame, in the case of bearings which are rigidly mounted on the shaft and in the frame and which prevent any compensating relative motion between shaft and frame. Both the mill motor layouts and that for the general purpose type motor provide against any damage from this cause.

Proper Fit of Bearing on Shaft and in Bore.

The application of roller bearings in all sorts of applications during a considerable number of years, has convinced us that the best way to secure the inner race of the bearing to the shaft is by making it a press fit on the shaft. This method eliminates all necessity for threading the shaft, lock nuts, dowel pins, holding screws, etc., and requires only the observance of the four following conditions:

1. The inside diameter of the race must be held to reasonably close limits, and the shaft itself must be ground with a reasonable degree of accuracy, so that there will be the proper amount of excess metal

on the shaft to make certain of a secure fit. The inner races of Hyatt Bearings are so manufactured, and the shaft limits which we recommend for their use are so proportioned that, taking the mid-point of the limits in each case, the shaft diameter exceeds the inside diameter of the race by an amount varying between .0006 in. for the small sizes to .0016 in. in sizes for shaft diameters about 3 in.

2. The race must be sufficiently thick so that its resistance to the stretching effect of the press fit will be sufficient to give it a firm hold on the shaft. It is evident that a very thin race could be pressed on and removed from a shaft which was .001 in. over size, with much less force than a thicker race, simply because the thin race would stretch enough to accommodate the excess metal on the shaft.

3. The bearing must be constructed with sufficient internal clearance to allow for the expansion of the race which is certain to result from the press fit.

4. The race must be sufficiently long so that the area in contact between the race and the shaft is sufficiently large to withstand the peening action which results especially from shock and vibration loads.

All of these four requirements have been fully considered and taken care of in the design of the Hyatt Bearing. The races are long, accurately machined and heavily constructed, and there is ample provision for expansion of the race caused by the press fit. Peening action on the shaft material is avoided not only by making the race long, but this effect is still further reduced by the resilient qualities of the Hyatt rollers which absorb shock loads and vibrations instead of transmitting them undiminished to the shaft.

The inner race need not be pressed into place, but can be expanded by heating in oil or water to about 200 deg. F. and slipping the race while hot into its proper position on the shaft. The expansion secured by heating the race in this way is generally more than the amount of excess metal required for the press fit; consequently, the hot race can be slipped onto the shaft with practically no effort and when it cools it will have as tight a hold as though it were pressed on. In using this process, care should be taken that a tempering oil or other high temperature oil is not used for the heating process, unless a therometer is employed at the same time, since it is easy to heat the oil to a point where the temper of the metal is drawn.

The demands of the Hyatt outer race are very simple. All that is required is a straight cylindrical bore, machined to reasonably accurate limits, into which the outer race is a push fit. This race is not intended to turn or creep in service, since the unit pressures at which Hyatt Bearings operate are so low that the elastic limit of the material is never approached. Consequently, it is not necessary to provide for any creeping of the outer race in order to present new surfaces to the load.

Operating Conditions of Motor Affect Design of Bearing Mounting.

All of the foregoing discussions have dealt with the requirements of the bearing itself. There are considerations of motor operation which have an equally important influence on the design of the bearing housing. Since the operating conditions of the various types of motors differ so widely, it seems best to consider them separately.

Mill Type Motors.

In addition to unusual ruggedness of construction, mill motors are expected to fulfill the following requirements:

1. The rotor and bearings must be easily removable from the lower half of the frame after the top half of the frame has been removed. Fig. 5 shows in a diagrammatic way that with a Hyatt equipped mill motor this process is identical with the method used with existing plain bearing motors. An eyebolt is

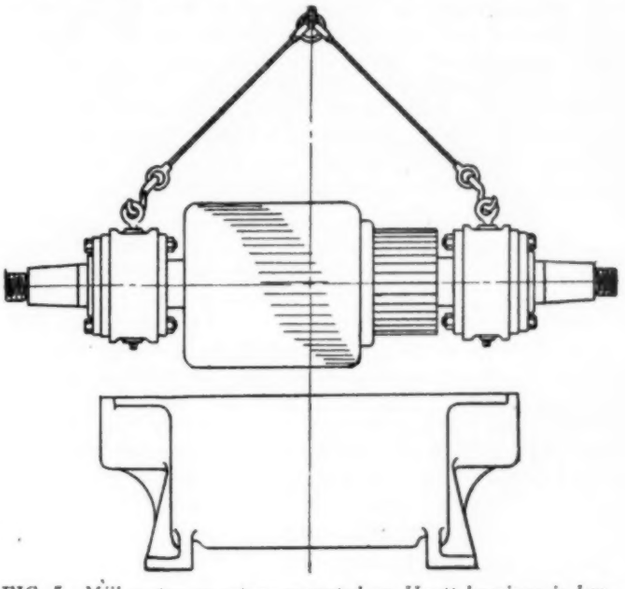

FIG. 5—*Mill motor armature mounted on Hyatt bearings is handled in exactly the same manner as plain bearing armature.*

screwed into the top of each bearing housing, and the crane sling is hooked into them. As far as the operation of changing the armature in a mill motor is concerned, the millwright gang doing the work would not even realize that they were handling anything other than a plain bearing motor, and there would be no need for any special treatment of any kind.

2. When putting the new armature into place in the motor frame, the same process is reversed. There is no need for any adjustment of any kind, either for the benefit of the bearings, or to regulate the end float of the armature. The armature and bearings are simply lowered into place in the frame so that the shoulders on the bearing housings fit in their place between the flanges on the lower half of the frame, and everything is ready for the top half of the frame.

3. When the armature and bearings are delivered to the electrical repair-shop, it should be possible for an ordinary mechanic without any special training with anti-friction bearings, to remove the bearing and housing from the armature shaft. The housings shown in Figs. 4 and 7 can be slid off the shaft without even the use of a wrench—simply pull the entire bearing housing complete with thrust collar, etc., from the shaft, leaving behind the inner race and the spacing ring. The housing complete with bearing can also be slid into place on a spare armature shaft without the use of any tools.

4. Inasmuch as the inner races for a given size bearing are interchangeable, it is evident that spare armature shafts could be stored with only the inner

race and spacing ring in position on the shaft, and when they were to be put in use, the housing complete with roller bearing could be slipped into place in a minute's time. In this way, six spare armatures could be taken care of by two sets of bearing housings without any risk of undue delay when the armatures were needed in an emergency. Such an arrangement would necessitate only the extra inner races and spacer rings.

5. Figs. 1 and 4 show a cross section of a bearing housing for a mill type motor which embodies all the requirements of the bearing, as well as the service demands of the motor. A study of these figures will show how the proper proportions of bearing and shaft have been observed, how alignment has been secured, lubrication facilities provided, how grit is prevented from entering the bearing, how lubricant is kept from escaping and the manner in which the armature is kept in its proper axial position.

6. In these designs advantage has been taken of the convenient arrangement of parts to provide further insurance of the roller bearing against prolonged neglect of the lubrication. The thrust collars have been enlarged in diameter to form an oil sling, which dips into the lubricant and throws it against the top and sides of the reservoir chamber, from here it runs through cored passages into the enclosure which houses the roller bearing. In this way the oil level can drop to half or three quarters of an inch below the level of the filler plug before either the roller bearing or the thrust surfaces were in any danger of running dry.

General Purpose Motors.

The demands made on general purpose motors require on the one hand greater latitude of application and wider range of suitability than mill motors, and on the other, less in the nature of special qualifications to meet specific conditions. The more moderate character of the duty imposed on general purpose motors reduces the need for facility of assembly and ease of inspection, repair and replacement. The entire design of the usual types of general purpose motors is made with the distinction in mind, and in laying out mountings for roller bearings, the same ideas have been incorporated.

Fig. 2 shows a typical Hyatt design for a general purpose motor. All of the specific requirements of the bearings as outlined above have been included, and the general case of assembly of the plain bearing motor has been retained. The end shield at the pulley end can be removed complete with the bearing as a unit, and slid off the shaft without disassembling the bearing housing by simply taking out the cap screws which hold the end shield to the frames. The inner race remains in place on the shaft, of which it has for all practical purposes become an integral part.

The locating collar, fastened to the shaft at the front end can be taken off if desired by removing the end cap and locknut. When this is done the end shield can be slipped off the shaft, taking with it the bearing except the inner race.

Judged by the service it performs, the front end bearing can generally be one or two sizes smaller than that at the pulley end. In Fig. 2 the same size bearing has been shown at both ends, in order to make both end shields identical. Some general purpose motors have shaft extensions at both ends, in which case either bearing must be suited to carry the pulley end load. When both end shields with their bearings are identical, the shaft design for the motor with one closed end is the same as the shaft for the same size motor with two extensions, except for the omission of one extension. Where the front and rear end bearings are different, the closed end motor must have a different shaft contour than the one with two extensions.

The decision in this respect rests primarily with the motor manufacturers, and is largely a question of the cost of carrying stocks and tool equipment for two sets of shafts and end shields as compared with the difference in cost of the larger and smaller bearings. Where conditions make different sizes of bearings advisable at the front and rear end, it can generally be arranged to use the rear end bearing of one size frame for the front end of a smaller frame, thereby keeping the total number of sizes down to practically the same figure as though both ends of each motor used the same size bearing.

Standardization of Bearing Sizes.

The general purpose motor, due to the large number of sizes in which it is made, and the variations in design involving closed ends, number of shafts extensions, etc., presents the most advantageous field for the standardization of bearing sizes.

It has appeared to us that any standardization program should take into consideration not only the reduction of the number of sizes to a minimum, which is, of course, the prime interest of users of large numbers of motors, but also the effect which the individual bearings in the standard series will have on the design of the particular size motor for which they are recommended. In order to secure sufficient bearing capacity, it should not be necessary to step up the shaft diameter under the bearing to a point where it produces a very sudden and undesirable increase in section over the standard shaft extension diameter, and where it may require redesign and enlargement of that part of the shaft between the bearings. Neither should the outside diameter of the bearing be out of proportion to the diameter of the motor frame and possibly reduce or eliminate the ventilating ports.

If a bearing of the 300 series is selected of the nearest size to the present plain bearing diameter, no changes are necessary to the shaft between the bearings nor to the shaft extension, and no undesirably abrupt changes in section of the shaft are introduced. The outside diameter of such a bearing is always in good proportion to the motor dimensions in general, usually requiring a housing not much larger in diameter than required by a plain bearing provided with a lubricant reservoir for ring oiling.

The load capacity of roller bearings makes it possible to recommend the 300 Series Bearings for motor service, using in each case a size close to the plain bearing diameter. The use of 300 Series Bearings for general purpose motors makes it possible to retain practically all of the present motor design intact with the exception of the parts immediately adjacent to the bearing.

Crane and Hoist Motors.

Fig. 3 illustrates a typical crane and hoist motor layout. The housing is integral with the motor head and the entire unit is easily removable from the shaft

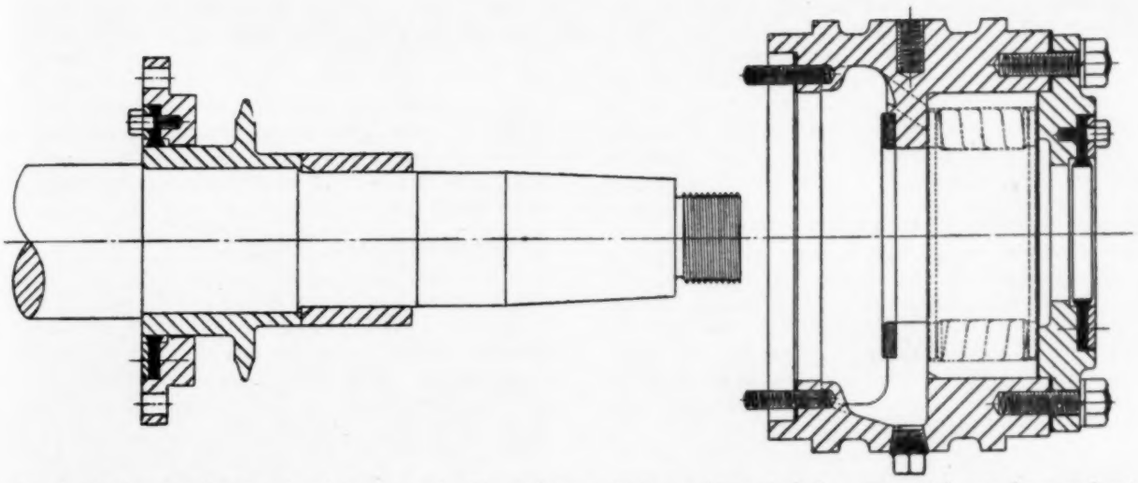

FIG. 6—*Mill motor bearing housing shown in Fig.* 1 *being withdrawn from shaft by removal of nuts from end cap studs. Housings shown in Fig.* 4 *and* 7 *can be withdrawn from hte shaft by hand without use of any tools.*

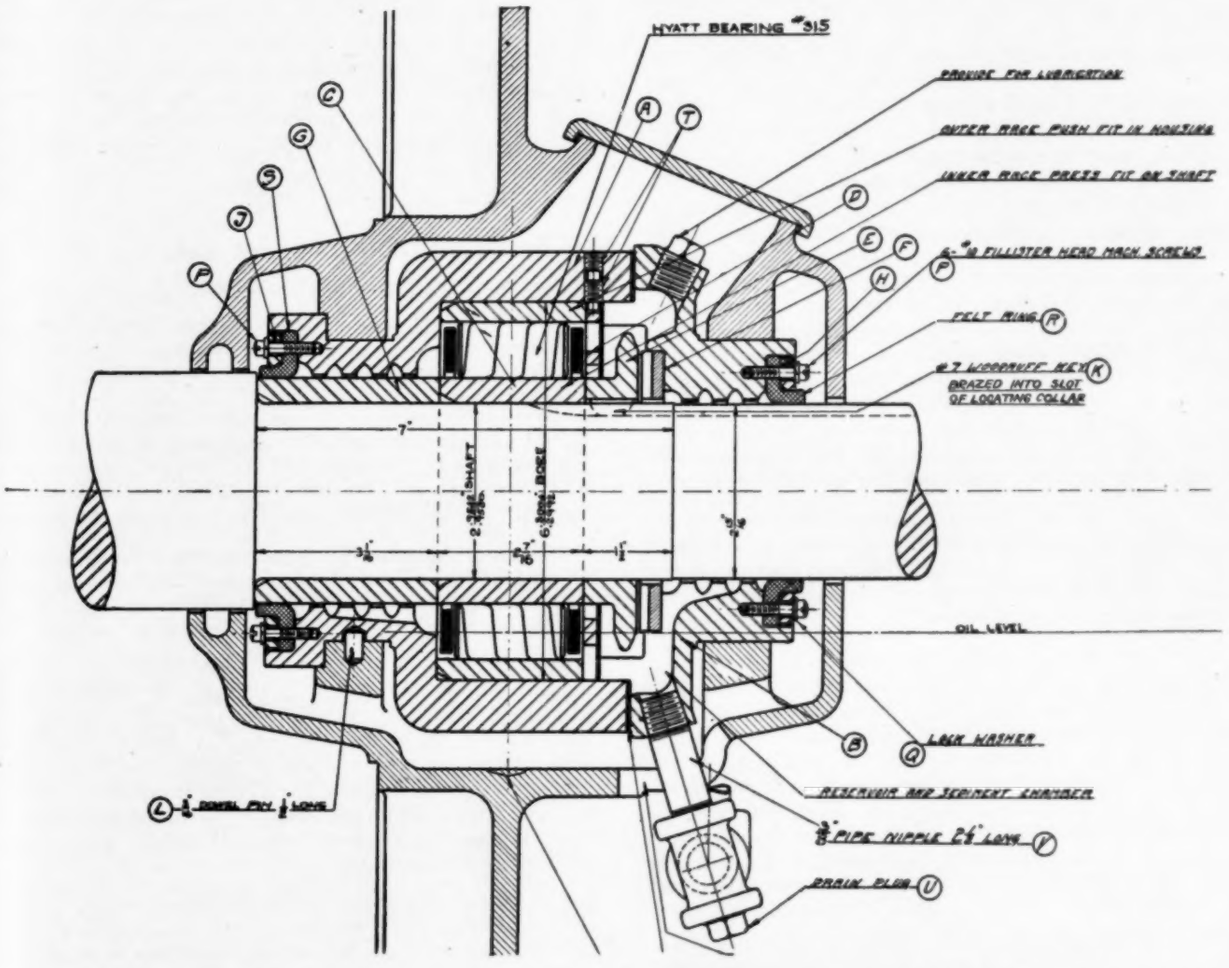

FIG. 7—*Changeover design for split end shield induction mill motor in cement mill work. Not grease saturated felt rings.*

and the frame by taking out the cap screws in the flange. Location of the shaft endwise is accomplished by means of a collar in each housing as shown, each housing taking thrust in one direction. All of the details of lubrication enclosure, etc., are worked out in line with the principles which have governed the design of mill motor and general purpose motor bearing housings.

Maintenance of Roller Bearings in Electric Motor Service.

The amount of time and money entering into the maintenance of roller bearings can be said to be inversely proportional to the thought and care put into the selection and mounting of the bearing. If bearings of adequate capacity are mounted in accordance with the principles outlined above, so that not only the particular requirements of the bearing, but also the needs of the motor user are kept in mind, we are convinced that they will render almost indefinite service.

Because of the widely divergent service which different motors are called upon to render, and the various degrees of attention which they receive, it is difficult to place even an approximate value upon the "expectation of life" of roller bearings in this class of work. Certainly five years can be taken as a minimum figure for practically any kind of service, since we have hundreds of motors which have run six and seven years in very severe applications.

One specific example is the motor in the coal cutters manufactured by the Sullivan Machinery Company, which have been equipped with Hyatt Bearings for the past 10 years or more. These motors operate at 1200 r.p.m., and impose a steady load of 870 lbs. per bearing, which is frequently raised to 1800 lbs. by the shocks and impacts of this unusually severe application. Although authentic records are not available, we are informed that some of these have run six or seven years on the original bearings and are still going. It would be difficult to imagine any more trying conditions, not only from the standpoint of shock and vibration loads, coupled with a fairly high speed, but also due to the fact that lubrication is very irregular, and that when the bearings are given some attention, they are lubricated with the black strap which is used almost universally in coal mines. Furthermore, the conditions of grit and corrosion are unusually serious.

In cases where operating conditions are more favorable, and where damage due to abuse and neglect is not to be expected in the same degree, there is every reason to believe the bearings will operate for 10 years or more without replacement.

The only attention they require is an occasional replenishment of the lubricant, and the flushing out of the bearing housing about every year or so, depending upon the amount of grit which is present. Aside from the advantages from an operating standpoint that can be gained by the use of a liquid lubricant in place of one of a more solid consistency, the only specifications for the lubricant are that it must be non-corrosive and have a flash point and a freezing point which will suit it for the operating conditions of the motor.

With a properly designed housing, holding a reasonable supply of lubricant, it should not be necessary to replenish the lubricant oftener than three or four times a year. Where plain bearing motors are taken care of by the same oiler who looks after those equipped with roller bearings, it is sometimes difficult to prevent the workmen from lubricating the roller bearing motors just as frequently as the plain bearing ones. If the lubricating arrangements are laid out as in the various layouts shown, this process will do no harm, since it is impossible to over-lubricate the bearings.

Flushing Out the Housing.

It is advisable about once a year to clean out of the housing any accumulated grit or particles of metal, and to dissolve any gummed oil, since the effect of this foreign matter upon the rate of wear of the bearing is cumulative. The simplest way in which the housing can be cleaned out is to remove the top and bottom plug from the housing, and drain out the lubricant. Then replace the bottom plug and pour kerosene or gasoline or a light lubricating oil heated to about 200 deg. F. into the top hole until it appears at the oil filler hole level. All plugs should then be replaced and the motor turned over slowly by hand a few times to distribute the solvent. After about 10 minutes the kerosene can be drained out and the motor turned over slowly by hand with all of the plugs removed, so that most of the solvent is removed. The bottom plug should then be replaced, and about a half pint of clean oil poured into the housing and the motor given a few turns to distribute it. This oil will remove most of the kerosene which might otherwise have a harmful effect on the lubricating qualities of the new supply of oil. When this small quantity of oil has been removed, and the bottom plug replaced, the housing is ready to be filled up to the proper oil level, and operation resumed.

Selection of Equipment Safe for Short Circuit Current

We expect to publish in a future issue a most interesting and practical paper on the "Selection of Equipment, Safe for Short Circuit Current".

A sudden short circuit tends to force an enormous initial current through all equipment in its path from the generators. Mechanical and thermal stresses vary as the square of this current and the results may be very disastrous if equipment is not properly applied.

During the past five years many articles have appeared in the technical papers and the proceedings of engineering societies, dealing with particular phases of this problem.

There is such an alarming discrepancy in the formulae which have been presented at various times, that one is much inclined to pass up looking into the matter carefully and to play safe by purchasing equipment which may be much more rugged and expensive than required. There is just as great a danger that equipment installed without proper study may be too weak for the stresses it may some day be called upon to stand.

It is the intention of this paper to bring together the essential facts of interest to the application engineer and to show by actual examples how they affect the selection of such equipment as bus supports, disconnecting switches, instrument transformers, etc.

Design and Application of Ball Bearings*

By T. D. CROW†

IN the development of the ball bearing, there has been a continual effort on the part of engineers towards reducing friction, with the result that the modern ball bearing is 99 per cent efficient and practically wearless. In view of the claims made that in ball bearings wear practically does not exist, the question is often raised—what then is the mark in the ball race which is noticed in a bearing that has seen even slight service? This mark is due to the planishing action between the balls and raceways which tends to level out the very slight surface inequalities which must exist, and in doing so causes a change in the surface finish which is noticeable to the eye in comparison with the original high polish. However, even with high powered microscopes and the most accurate of measuring and weighing instruments is it possible to determine the amount of wear or depth of this mark even on bearings subject to 100 per cent overload.

Very slight frictional losses in a ball bearing are due to two reasons—First, the greatest loss is that the friction between the balls and separator, or ball spacing cage, is sliding friction, not rolling friction which exists between the balls and races. The second cause is that the races and balls slightly deform under load

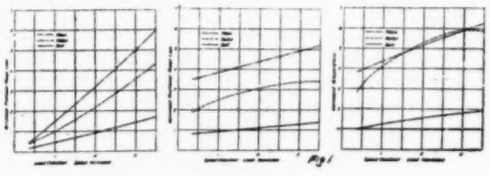

FIG. 1.

and power is absorbed by the material in returning to its original shape. However, both these losses are extremely small, with the result that the frictional loss in a ball is practically constant and independent of speed and load, so that rise in temperature under operating conditions practically does not exist. These losses are diagramatically shown in Fig. 1.

Regarding the carrying capacity of a ball bearing, the outside diameter, bore and width have become internationally standardized and we therefore have a definite cross-section between the inner and outside diameters in which to insert a load carrying means, and as the capacity of the bearing is proportional to the number, diameter of the balls and relation of the race cross-section curvature through the ball diameter, it is desirable to use as large a diameter and as great a number of balls as possible and yet, maintain race thickness of sufficient strength to prevent excessive deflection of the races between the balls.

In a radially loaded bearing, practically only one-third the number of balls in the bearing support the load. That is, as shown in Fig. 2, the balls below the centerline of the bearing at right angles to the load application are actually free of the inner, or load resisting race. This is due to the slight initial radial clearance being increased by the deformation of the balls and that section of the races which are resisting the load. In an angular contact bearing, or one which is sustaining both radial and thrust loads, a larger number of balls resist the load. With a maximum radial and zero thrust load, the resultant load will

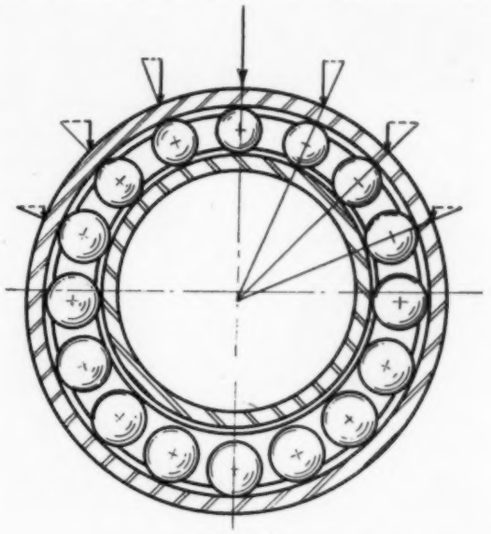

FIG. 2.

be radial and as before mentioned, only those balls in the upper half of the bearing are in a position to support the load. As the relative magnitude of the radial and thrust loads vary, the resultant load changes direction so that more of the balls are in a position to assume a portion of the load. Finally, when the condition of zero radial and maximum thrust is reached, each ball assumes an equal portion of the load and the maximum capacity of the bearing is obtained.

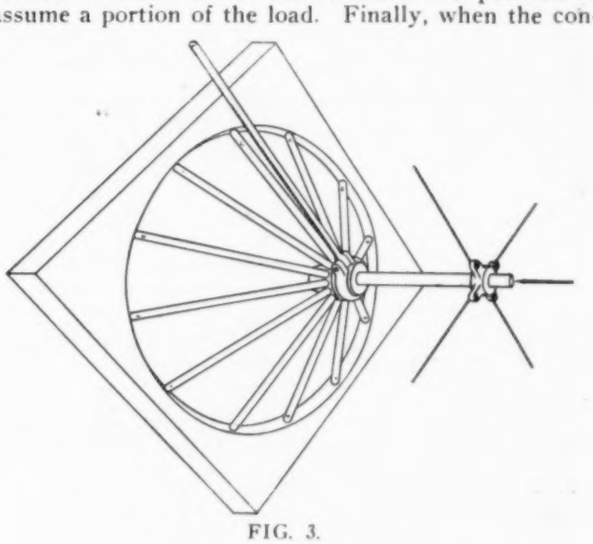

FIG. 3.

Fig. 3 diagrammatically shows the effect of thrust and radial load, the spokes of the wheel representing the balls, the hub—the inner race. Under pure thrust

*Presented at Philadelphia Section, January 8, 1924.

†Chief Engineer, The New Departure Mfg. Company, Bristol, Conn.

load, or a load in the direction of the arrow, all the spokes are equally loaded. Now, if we apply a radial load by means of the tie-rod, the load on spokes 4, 5, 6, 7, 8 and 9 will be reduced, and the load on spokes 1, 2, 3, 10, 11 and 12 increased. Further application of radial load eventually would cause the first mentioned spokes to fall out and the thrust and radial load would be supported by spokes 1, 2, 3, 10, 11 and 12.

As previously mentioned, in order to obtain the maximum carrying capacity for a given ball diameter

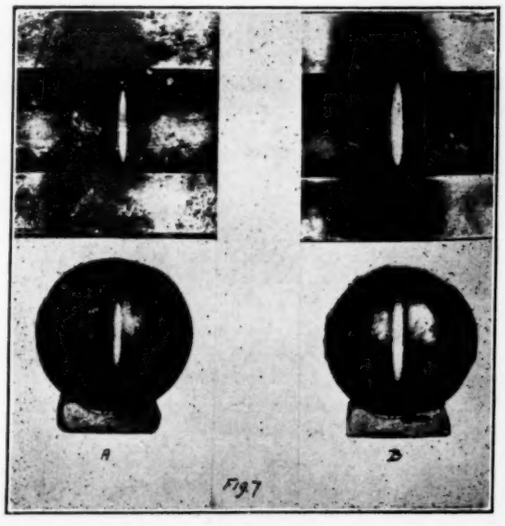

FIG. 4.

and still keep the frictional loss at a minimum, the race cross-section curvature is made slightly larger than the diameter of the ball and under the application of load, deformation occurs which slightly flattens these curvatures out until the ball is changed to a barrel-shape at the load contact areas.

As the speed at the surface of the ball is greater at the center or maximum stressed area, than at the extreme ends of the contact area, a slight slippage must occur over this area. The slippage, however, is so slight that it can only be determined under a very excessive overload laboratory investigation. It is obvious that if the curvature of the race was equal to that of the ball, minimum deflection would occur and the maximum amount of slippage would result due to the greater variation in surface speed and the contact areas. If, however, the curvature is made too large, greater deflection of the raceways and balls will occur with resultant overstressing of the material and consequent reduction in life and carrying capacity.

The first thought is that if an equal surface speed exists, the ball will spin and slide. This, however, is not the case as the slippage at one end of the contact area is equal to the slippage at the other and therefore, there is no force tending to spin the ball. This leaves us with the thought that as the ball only rotates on one axis, wear must occur. However, it must be borne in mind that the balls below the centerline of the load application are not under load and therefore, change their axis of rotation due to slight vibration of rotating parts. It is possible to apply pure thrust load so that the balls are equally loaded and they then continually rotate on the same axis. Experimentally, a ball was produced. This ball is a very excellent example of the so-called slip bands and was produced in experimentally testing an angular contact bearing under twice its rated thrust capacity, it being, as before mentioned, impossible to produce this excessive slippage under normal operating conditions. At this point it is well to mention that the load supporting area in a ball bearing is of an appreciable size and not a point contact.

Fig. 4 is a photograph of the actual contact area between ball and outer race obtained when applying the maximum load that the ball is called upon to sustain.

In mounting ball bearings we are continually advised that a press-fit must be used on the rotating member — otherwise slippage will occur with resulting reduction of the shaft diameter. The result is that generally excessive press fits are used and these may have a very detrimental effect. The inner race of a bearing, which is generally the member that is mounted with a press-fit, has relatively a thin section when compared to the shaft diameter and therefore such expansion or contraction of parts which occurs when pressing a piece on a shaft causes expansion of the inner race. Now, as the radial clearance in a ball bearing is practically zero, this clearance can be absorbed by the press fit so that actually an initial load will be put on the bearing which not only makes the bearing noisy, but will also reduce its available life. As torque is not transmitted through the bearing, there is no need for an excessive press fit—only sufficient pressure should be used to keep the surfaces in contact under the slight deformation which must occur under load. There is no need to increase

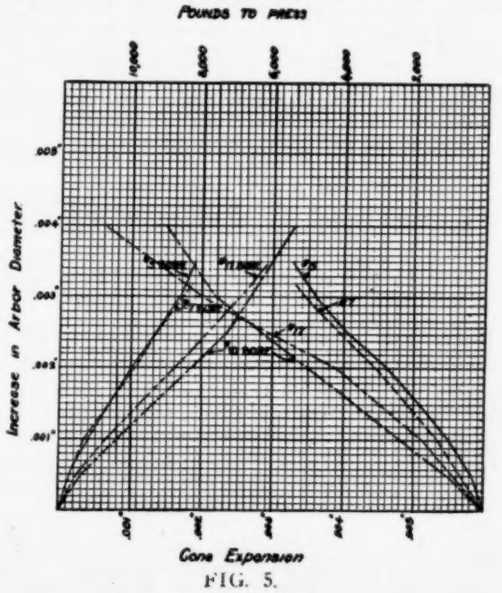

FIG. 5.

the amount of press fit with the bearing size as there is sufficient increase of surfaces in contact to sustain the load. Again, it must be remembered that the larger the bearing the thinner the race section in proportion to the shaft diameter and therefore, there will be a greater expansion for a given amount of press fit.

Fig. 5 shows a chart illustrating the effect of various press fits on five, seven, 10 and 17 bore bearings. The chart also shows the pounds pressure required to

obtain these press fits on a hardened shaft slightly lubricated. It will be observed that the average expansion is approximately 50 per cent of the fit. However, it should be noted that the minimum expansion for a given amount of press is obtained on the small bore and the maximum on the larger bores.

If a bearing for constructional reasons has to be mounted with a loose fit on the shaft, the fit should be made as tight as possible; otherwise, no locking means used can prevent rotation and it is also necessary that the direction of the thread of the locking member be such that it will tighten up under load or any locking means used to prevent rotation of the nut will be sheared off.

It can readily be seen that if the shaft circumference is smaller than the circumference of the bearing bore, the section of rotation of the bearing and shaft will not coincide and therefore, there exists a very high reduction internal friction drive and it is obvious that any locking means which has such a small frictional surface as is presented between the locknuts and the race face could not resist this turning motion. At this point would like to mention precaution against the use of a split locknut.

Tightening screw in this case actually relieves the pressure on the threads in the nut and therefore is working against itself, but if the screw is applied, it is obvious that pressure on the threads is increased when tightening the lock screw. We hardly need mention the well-known fact that the non-rotating race should be a light push fit in order that it may slowly creep under the load vibration, so presenting a different portion of the race section to load. However, in the case of high speed bearings it is perhaps inadvisable to use this condition because the creep of the outer race is so materially increased that wear of the supporting member may occur. If found necessary to use a loose fit for constructional reasons, the housing can be slotted and flat springs inserted which will

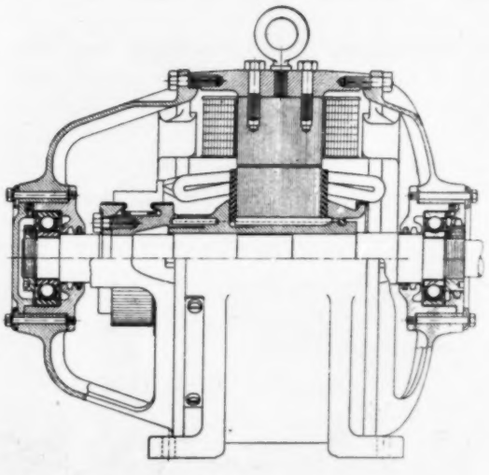

FIG. 6.

slightly increase the friction and so slow down the speed of rotation and yet, the bearing will be free to move endwise and accommodate machining inequalities.

Regarding high speed installations, that is, speeds above 4,000 r.p.m., experience has proven that the angular contact bearing in this case gives the most satisfactory service. If a straight annular bearing is used, failure first occurs on the separator due to the amount of work which has to be done in pushing the balls around in the unloaded section, it being obvious that as the straight radial or annular bearing has only one-third of the balls carrying the load or driving the separator, there are two-thirds of the balls on which the separator has to transmit force, whereas in the angular contact bearing, if sufficient end thrust is used, the separator is practically relieved of all its load

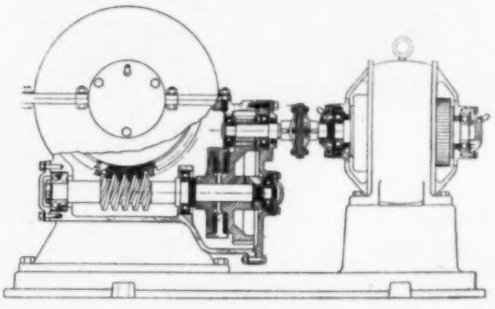

FIG. 7.

In applying ball bearings to electric motors, experience so far has shown that in order to obtain the size of the bearing, the load should be computed as belt loads which can run very high when compared to those obtained from gears. The fact that electric motors may run continuously day and night does not have such a bearing on the size as would appear at first thought. The fatigue of the material in a ball bearing is not the same as in any other type of machinery. Our own tests so far have shown that a bearing which is run continuously day and night does not fail any quicker than a bearing which is run for 10 hours, and then rests for 12. These tests were run under twice the bearings rated capacity. There does not seem to be much leeway in designing a mounting; however, we believe that the mounting should be so designed that if possible, the armature and bearings can be removed as a unit, leaving the bearings fully protected. This is illustrated in Fig. 6. The bearing container should be so designed that the bearing can be removed from the shaft without causing pressure to be exerted on the outer race. Also, that relation of the closure to the housing should be such that when the closure is removed, access to the locknuts is readily obtained without offset or box wrenches.

Regarding the seal, as the general practice is to use grease for speeds below 4,000, feel that a plain grease groove and internal drain supplemented by a felt groove on the outside to exclude dust is, perhaps, the most satisfactory and simplest closure method. However, particular care should be taken that the volume of the housing is larger at the outside diameter in order that the lubricant which is churned up by the bearing can be thrown out and so relieve the pressure from the junction of the closure and the shaft.

Fig 7 is another application which is, perhaps, cheaper but not as satisfactory inasmuch as the bearings have to be removed from the shaft before access to the armature can be obtained and so, there is every possibility of the press fit of the inner race on the shaft being destroyed.

The mounting shown in Fig. 8 overcomes this objection but allows the bearing to be exposed to dirt while the armature is being handled.

Fig 9 shows the application of ball bearings to a 2-hp., 200-cycle induction motor which operates at speeds as high as 12,000 r.p.m. with excellent success. It will be noticed that two angular contact bearings, opposed to each other, are mounted at the front or driving end, and a single angular contact bearing,

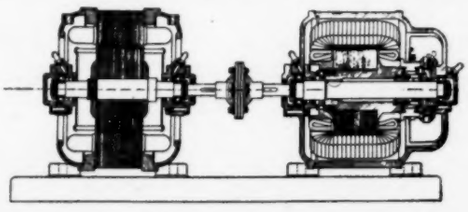

FIG. 8.

backed up by a spring, is used at the other end In operating at these speeds particular care must be taken that the amount of lubricant is kept to a minimum—otherwise, excessive heating will occur through the churning up of the lubricant. Our practice is to feed three or four drops of oil every two hours. In our own plant we have motors of a similar type. Though slightly smaller, operating at speeds as high as 20,000 r.p.m. with complete satisfaction.

In conclusion, would like to state that we believe a ball bearing electric motor should be assembled so that when mounted on a shaft the bearing is under very slight tension. This may produce a slight hum due to vibration of the exceedingly hard metal, but if this is not done a much greater vibration will be set

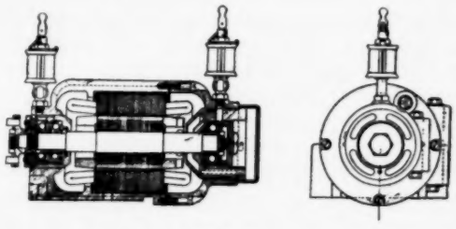

FIG. 9.

up due to the fact that the electrical and mechanical centers do not always coincide, which causes a vibration as each armature winding leaves the pole piece. Also, in a motor which is being constantly stopped and started, if the bearing parts are not under tension, there is a tendency for the unloaded balls to slide due to the very high acceleration which is sometimes present in small electric motors and therefore prevents the unloaded balls from gradually obtaining their rotative speed.

Discussion on Ball and Roller Bearings on Pages 256 to 278

The Magnetized Roller Conveyor

By HARRY HOLLES*

THE theory of the magnetic roller type of conveyor is that of utilizing the tractive forces of the electro-magnet as effective to increase the coefficient of friction between the surface of the roller and the bar being conveyed. We have found that this increase of friction enables us to do the same work with one magnetized roller that would require six rollers of the unmagnetized type.

The magnetic unit of the conveyor, as used on three merchant mills at the Lebanon plant of the Bethlehem Steel Company to convey the bars from the cooling bed to the shear consists of two wrought-iron rollers, four inches in diameter, magnetized by an electro-magnet.

Fig. 1 is a photograph of the first magnetic roller unit as designed and constructed at the Lebanon Plant. This unit as shown is driven by a 3-inch belt. This unit is equipped with a loose pulley, the belt being shifted by means of a magnet as shown, and operated by the shearman. This unit was one of two units installed on a 100-ft. conveyor at the Lebanon plant in February, 1922, and in the two years' operation 22,680 tons of iron bar have been conveyed to the shear. Up to the present time, no electrical or mechanical parts of this magnetic unit have been replaced, and the gear maintenance. generally met with in the mechanical type of conveyor, has been eliminated.

Fig. 2 shows a plan and elevation of our standard magnetic unit. The base plate has been extended to take care of the mounting of a belt-shifter magnet, or an electric motor for operating the rollers. The pedestal type of bearings for the magnetized rollers are composed of non-magnetic material, as well as the

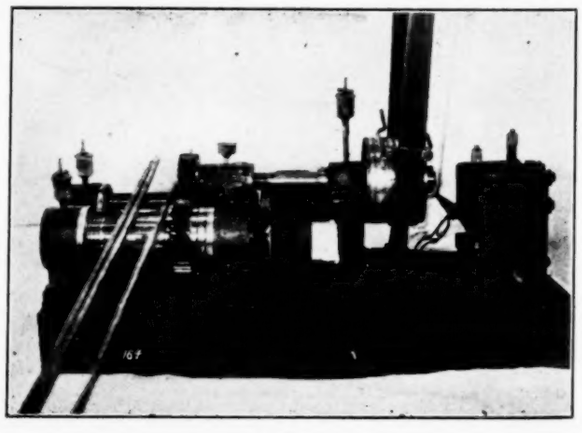

FIG. 1.

*Elec. Supt., Bethlehem Steel Co., Sparrows Point, Md.

gear wheels mounted on the rollers, to prevent short-circuiting the magnetic lines of force.

Fig. 3 shows, in the form of three curves, the result of a test to determine the ultimate or slipping pull of the magnetized rollers when conveying ¾-in. round bar. The rollers were driven by a 2-hp., 1-hr. mill rating, compound wound motor. Eight bars of ¾-in. round bolt iron were used, the bars making contact with both rollers and the pull of the bars being registered on a spring balance scale, the rpm. of rolls and input to driving motor being read simultaneously with the indication of the spring balance. You will note the first bar indicates a pull of 25 lbs. for one bar. The bars were added one at a time until a total

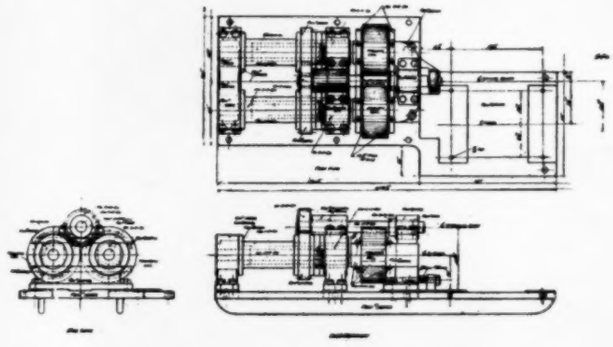

FIG. 2.

of eight bars were in place. From the "pounds pull" curve you can see that the capacity of rollers is reached with six bars, the pull being 102 lbs., or an average of 17 lbs. per bar. The curves of the hp. or drive and speed of rolls check this point, as it is evident by a consideration of the three curves that there is no change in the bar pull, hp. of drive, or roller speed with the addition of the two bars after the sixth bar.

To illustrate the few number of rollers required for the magnetized roller type as compared to a standard form of the mechanical type, we can take as an example a conveyor for finished bar, 150 ft. long. For a standard type of mechanical conveyor, we would have approximately 38 8-in. rollers, located on 4-ft. centers. To support and drive these rollers, 114 bearings and 76 spiral or miter gears would be required. A magnetized roller conveyor, to do the same work, would require six 4-in. rollers in sets of two rollers, each set of rollers being located on 50-ft. centers. The number of bearings required would be 18 and the number of gear wheels 15, which would be of the simple spur and pinion type.

With the smaller number of rollers to be accelerated, the hp. of the driving motor would be at least one-half that required by the mechanical type.

On our 10-in. central bar mill, we are using three sets of magnetic rollers on a conveyor 150 ft. long. Each set of rollers is driven by a 2-hp., 1-hr. mill rating, compound wound, d.c. motor. The motors are geared to the rollers to give a bar travel of about 250 ft. per minute, and on a 10-hour turn this mill has turned out approximately 100 tons of 19/32-in. spike iron, or an average of 10 tons of iron bar per hour, which the conveyor was called upon to handle. These bars are generally conveyed eight at a time, and no trouble has been experienced in keeping the cooling bed clear of iron.

Our practice indicates that the magnetic conveyors can be built and installed for about one-half of the cost of the mechanical type of conveyors.

It is the speaker's opinion that the magnetized

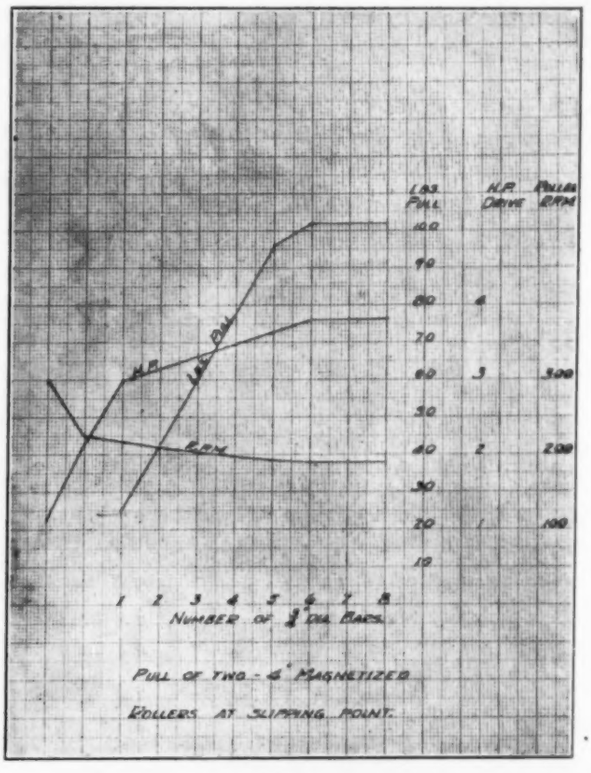

FIG. 3.

roller conveyor can be designed to handle almost any shape of iron or steel as it comes from the mill, and when the lower first cost, as well as the very low maintenance cost becomes more generally appreciated, we can look for a rapid development of this type of conveyor.

Iron & Steel Exposition

The leading manufacturers of equipment used in the Iron and Steel Industry will display their products at the Iron and Steel Exposition being held in connection with the annual convention of the A. I. & S. E. E.

Application blank and floor plan are shown on pages 202 and 203 of this issue.

If you have not already reserved space, you should do so at once by writing

John F. Kelly, *Chairman*,
IRON AND STEEL EXPOSITION,
706 Empire Building **Pittsburgh, Pa.**

Third Rail Locomotive for Yard Service

By F. O. SCHNURE*

THE subject scheduled for me to present tonight has been the topic of so much discussion in past meetings that I have simply outlined our late work in this direction, so that others might take advantage of this opportunity to lend us their experiences on yard electrification.

During the past five months we have had an interesting and reliable piece of equipment working in our plant that I wish to present to your attention so that my paper will therefore cover two subjects.

(a) Third Rail Locomotives for Yard Service.

Electric power for the various operations in a steel plant is recognized as having a distinct advantage over steam because of the high efficiency, the economy of operation and maintenance of the electric motor. There is, therefore, only one drawback to the replacement of steam locomotives by electric locomotives and that is the availability of energy at the point where work is to be done. The third rail, the overhead trolley, and the storage battery represent the present day means of bringing electric power to the locomotive. As all these methods are expensive to install and more of a nuisance to maintain than the

FIG. 1.

average mill distribution system, the motorization of the rails has not kept pace with the mills. Motor cars for special jobs, or transfer cars, coke transfer cars, ingot buggies, etc., are the rule, but the third rail has yet to be extended over the plant yards.

In the Maryland plant it is necessary to transport blooms a distance of 2,600 feet from the blooming to the rail mill, and a high speed motor car was built and 4,000 feet of third rail installed to perform this service. Following the standard practice of this plant, a 3x3-in. billet of special section was used as the rail, supported to give bottom contact and arranged to give the minimum of obstruction to general yard work. A motor car weighing 26 tons was built by equipping two standard 60,000-lb. trucks with four 50-hp. 38-B railway motors and is used in connection with an idler car bearing collector shoes, this arrangement being necessary to span gaps in the rail over roads and switches. Series parallel magnetic control with an electric reverser is used. The locomotive is provided with air brakes and electric heaters.

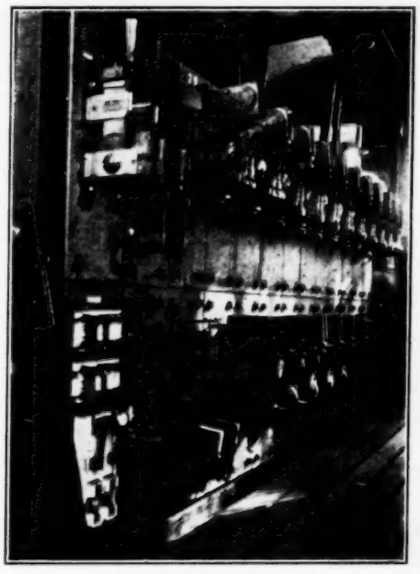

FIG. 2.

The usual practice is for the locomotive and idler car to handle a car with about 16 tons of blooms per trip. During the month under observation the locomotive traveled 644 miles, hauled 10,500 short tons of blooms and used 132 watts per gross ton mile. Comparison with our steam units in continuous service shows that this gave a fuel ratio of 44 per cent, allow-

FIG. 3.

*Elec. Supt., Bethlehem Steel Company, Sparrows Point, Md.

ing 2 lbs. of coal per kwh. This compares with 43.6 per cent obtained last year on a motor car pushing hot metal a distance of 1,900 feet from one open hearth to another. The total consumption for power, heat and lights amounted to 7,860 kwh., which at 1c per kwh. cost $78.60. The repairs were negligible, the operating labor considerably less than that for standard steam service, so that the actual cost of electric operation runs about 40 per cent of the previous steam operation.

The above locomotive has a free running speed of from 15 to 20 miles per hour, it being necessary to transfer the blooms with a minimum loss of heat. Its gear ratio and weight render it too light for the usual class of shifting found in steel plant yards. Future locomotives will be practically doubled in hp. and weight and the speed cut in half.

FIG. 4.

Load Regulator Used with Turbo-Generator

By F. O. SCHNURE*

THIS load regulator was developed to allow the operation of an industrial power plant in parallel with a central station so that in case of a fluctuating load the peaks can, (a) be absorbed by the industrial plant, (b) be absorbed by the central station, (c) be split between the two systems.

When two power systems are connected, the speed control must necessarily fall on the larger station, which in the present day of interconnection means that the burden of speed control falls on the central station. When an industry such as a steel plant has a power plant that is able to carry practically the whole load, and where there is not a power exchange clause in the power contract, there is a natural reluctance to cut the industrial plant back to the low part of the load curve so that the central station will get the block of power resulting from the fluctuating load. There is also a reluctance to absorb more of this varying load under penalty of "giving" power to the central station.

This device is best described by the accompanying diagram. The turbine governor valve control is so arranged that the load regulator motor can unload and load the turbine. The control of the motor is vested in a contact making wattmeter type relay placed on the incoming central station circuits.

Operation is as follows: Assuming the turbine partially loaded and the central station circuits with no load. The relay is in a state of equilibrium. Suppose the load picks up and is absorbed by the central station. The relay makes contact to raise the turbine and absorb the load until a state of equilibrium again exists. Suppose the load drops off, the wattmeter relay makes contact in the lower position until a balance again exists.

The charts shown will give a clear indication of the operation of this equipment. With a fluctuating load of 10,000 kw. it is possible to operate in parallel with a central station and neither buy nor give more than a couple hundred kwh.

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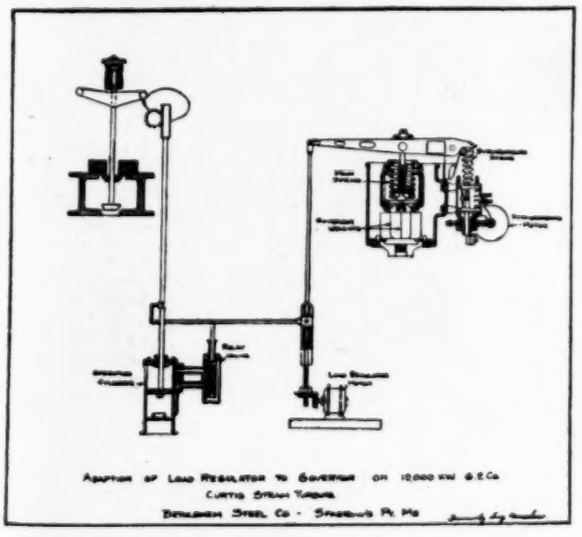

FIG. 1.

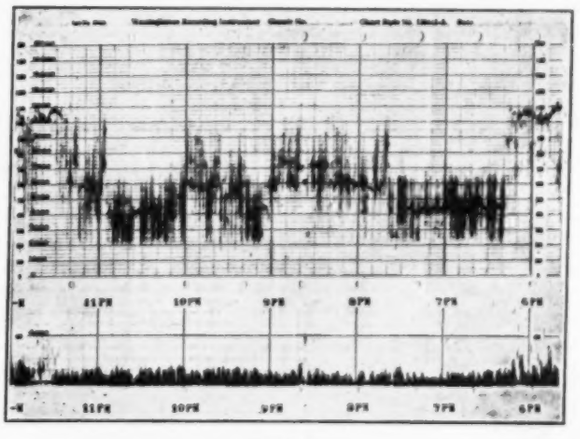

FIG. 2.

*Automatic Electric Stations for Steel Mills**

By CHESTER LICHTENBERG†

AUTOMATIC electric stations are relatively a new development in the art. The first one was installed in an interurban railway substation not quite ten years ago. Other installations soon followed. Their use was extended to all classes of electric railway, central station and industrial applications until today over 1,000 rotating electrical machines are operating under the control of automatic switching equipments. In this connection it is interesting to note that installations of automatics are almost doubling in number each successive year. It may also be astonishing to realize that at the present time the automatic stations which are in service represent an investment of machinery, buildings, land, etc. over thirty millions of dollars. Their success seems assured and they are fast becoming a potent factor in the rehabilitation of rapidly growing electricity supply networks and systems of transmission and distribution.

The automatic electric power station is one in which all of the functons of connecting the electrical apparatus to the feeding-in sources and supply mains, starting the electrical apparatus, guarding it against damage or destruction from any one of a number of hazards, and disconnecting it from the feeding-in and supply lines, is accomplished without the aid of an attendant in the station. The starting and stopping of the equipment may be in response to load demand as in interurban railway service. Or, it may be cyclic and controlled by a time clock as in certain industrial applications. Or, it might be in response to a push button or similar control near at hand or quite remote. No matter how the starting or stopping impulses may be given, the electrical apparatus in an automatic station is made to go through its performance without the aid of an attendant as long as power is supplied to it.

The advantages of automatic electric stations are numerous. Their precise evaluation for any installation depends largely on local conditions. The principal features, however, are reduction in operating costs, reduction of light load losses and reduction of feeder losses. They permit of improved responsivity to operating requirements including better voltage regulation and service continuity. An added feature, which is hard to comprehend until one has had experience, is in reduced maintenance charges. This may seem strange since there is more electrical apparatus in an automatic station than in an equivalent manual station. However, the experiences of a number of automatic station users vouches for the validity of this seemingly extraordinary statement.

The types of automatic stations vary with the service requirements and the apparatus used. Automatic switching equipments are now in successful operation on electric street railways in city service, electric interurban railways, electrified steam railroads, Edison 3-wire networks, 2300 volt a.c. distribution networks, 6600-73000 volt transmission and tie lines, mine haulage service, mine cutting operations, elevator control, battery charging, steam railway signal power control, paper novelty mills, hydro-electric developments, and steel mills.

Steel mill engineers were among the first to recognize the advantages of automatic stations. Witness, for example, the fact that the first completely automatic station was built by a steel mill and electric railway engineer from extra control parts supplied a large steel mill electrification. Also, the fact that some of the principal control features used in automatic stations today were developed by the experience obtained in steel mill electrification goes far to indicate the co-operation of the steel mill electrical engineer in the development of automatics.

The application of automatic switching equipments to steel mill electrification requires a thorough study of the electric power requirements and utilization in such establishments. Continuity of electricity supply is paramount for certain operations such as rolling mills. Relatively constant voltage is desirable for cranes, mill tables and the like. Good power factor reduces the line losses, and if the primary power is purchased, the power factor materially affects the rate on which payment for the electricity is based. Ability to successfully meet large power demands and to ride over momentary interruptions in primary supply are additional features which the development of automatic switching equipments has made possible and which will soon be required by steel mill engineers.

One class of steel mill electrical equipment which lends itself to automatic operation is the motor-generator set for furnishing the d.c. supply for the operation of various pieces of equipment in the mill. Sometimes these motor-generators are located in or near the mill power house. Other times, and more frequently, they are located in substations strategically placed in the mill yard. The motors of these sets are usually self-starting through compensators. They can readily be made automatic starting either in response to load requirements or, in anticipation of these requiremnets, by a central power director. The starting in either case would be accomplished as in any automatic motor-generator set. First, the a.c. current supply voltage would be checked to insure its being sufficient to start and run the set. Then, the supply service would be checked to be sure that 3-phase power was available and that the phase sequence was correct: Next all the motor-generator bearings would be tested to insure that they had not previously been overheated. Finally, the starting and magnetizing oil circuit breakers would be closed and the synchronous motor field would be disconnected from its usual exciting source and bridged by a field discharge resistor. This would be of sufficient value to insure pull-in torque with the minimum voltage at which starting could occur. Then, at about 95 per cent synchronous speed, the field discharge resistor would be removed and the field excited from the usual exciting source. The field transition would be made in two or three steps, depending on the size of the synchronous motor. Afterwards, the synchronous motor would be transferred from the starting to the running connections. Next, the voltage of the d.c. generator would be balanced with that of the d.c. bus and when these voltages agreed, the d.c. machine would be connected to the bus. Then, by automatic field regulation, the d.c. machine terminal voltage would be set for any desired value and from then on it would take its predetermined share of the mill load.

There are many variations of this simple operation which have been developed for automatic station service

*Presented at Cleveland Section, April 12, 1924.

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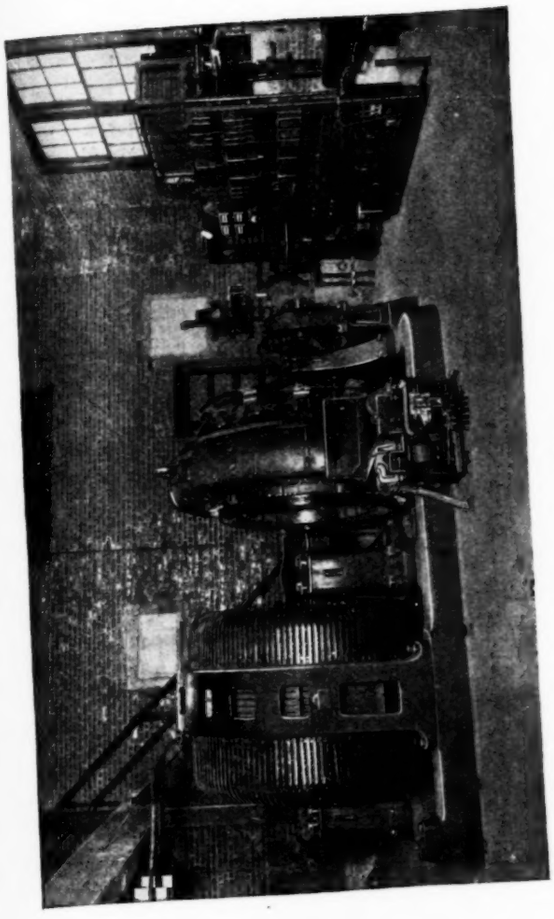

FIG. 2.

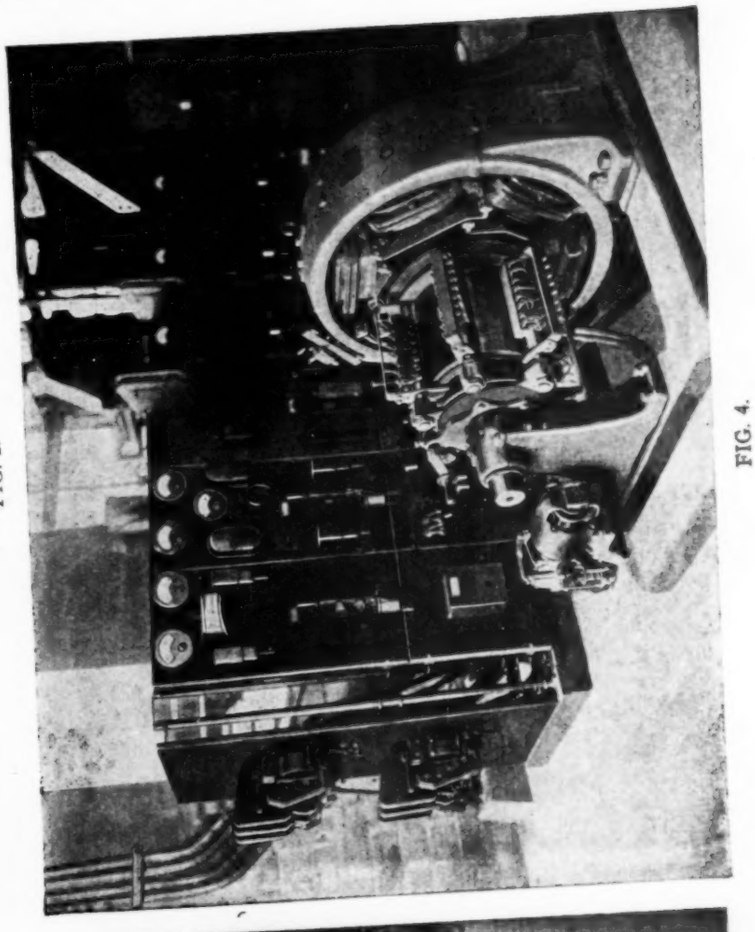

FIG. 4.

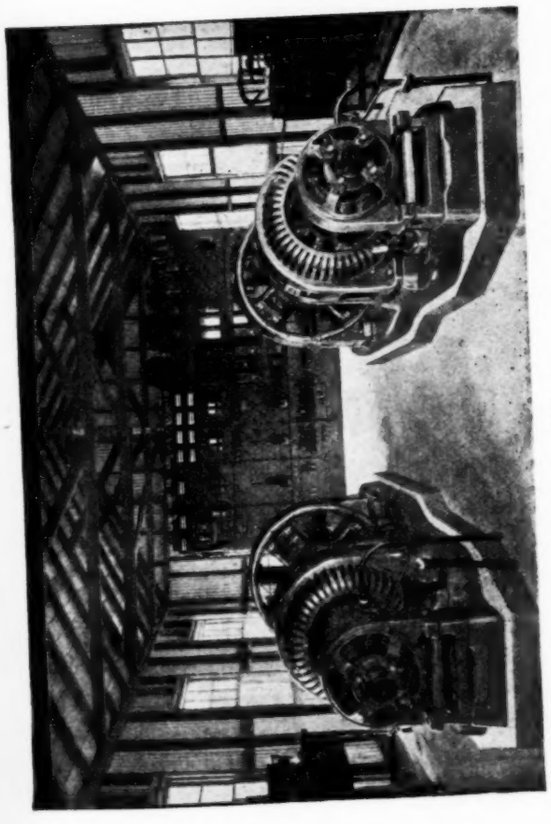

FIG. 1.

FIG. 3.

and which might well lend themselves to steel mill electricity supply. One such scheme contemplates the synchronous motor of the set so designed that it may be self-synchronizing after momentary interruptions of primary supply, if such interruptions were of sufficiently short duration and if the d.c. power requirements were not too great to permit the motor-generator to remain above 70 per cent synchronous speed. In such a design the interruptions of the a.c. supply would interrupt the synchronous motor field circuit and connect the compensator into circuit with the main a.c. breaker remaining closed. If the a.c. power interruption was of only a few seconds duration, its return would find the synchronous motor ready to receive it and the motor would go through the latter part of its starting sequence and promptly pull back into step. In the meantime, the d.c. service would not have been interrupted though the distribution voltage might have been reduced. This characteristic performance is particularly valuable where power is purchased or supplied over lines subject to lightning disturbances or other momentary interruptions.

Continuity of power supply in spite of disturbances on the d.c. network may be assured by suitable equipment for the d.c. service. One design uses a compound wound d.c. generator with the series field differentially connected. This field is usually by-passed by a circuit breaker which shunts out the greater part of the series current. In case of overload or short circuit the series field shunt is opened and the series field permitted to act differentially. In short circuit cases, it acts to make the generator a constant current machine with terminal voltage inversely proportional to the current flow. In the case of reverse current, the series field acts to assist the shunt field and prevent overspeed of the set. When the abnormal d.c. conditions are removed the series field shunt is automatically reclosed and the set again acts as a practically shunt wound machine. This design is well adapted for application to d.c. networks where the normal current increments are relatively moderate compared to the capacity of the motor generator set.

Another design has been perfected for use where the power increments are large and rapid when compared with the capacity of the motor-generator set. This design uses a shunt generator provided with an exciter and a counter e.m.f. set. The armature of the generator of the counter e.m.f. set is connected in the field circuit of the main generator. Its field circuit is varied to meet any predetermined set of conditions. In one example, which has been in operation well over a year, the main d.c. generator is arranged to maintain pratically constant terminal voltage under a load up to about 150 per cent normal. Beyond this point the design of the equipment is such that the terminal voltage is decreased inversely as the current draught increases so that in effect the machine becomes a constant current machine beyond 150 per cent normal current load. This design is also arranged so that momentary currents of three times normal or twice reversal will be safely controlled.

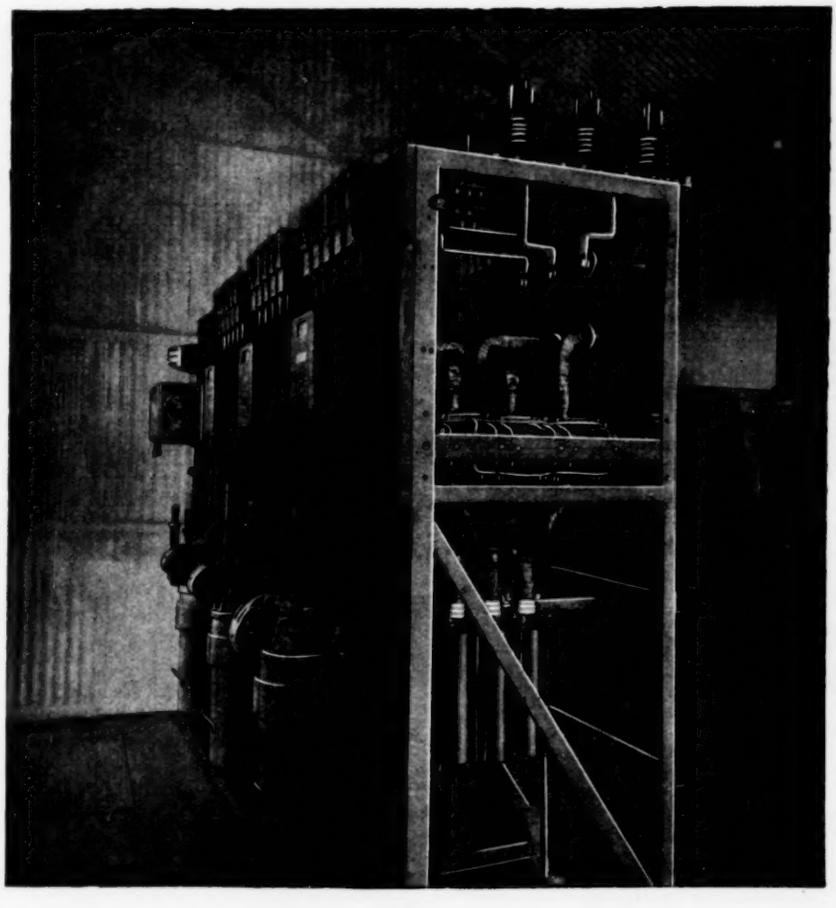

FIG. 5.

Synchronous condensers form another class of electrical machines for which automatic operation in steel mills is particularly well adapted. Synchronous condensers can now be advantageously located in the mill yard and can be used to correct low power factor conditions caused by the use of induction motors. Automatic synchronous condensers are started and placed in service in practically the same manner as synchronous motor generators. They are quite a bit simpler in their design since there is no d.c. machine to equalize on to the bus. Automatic synchronous condensers are usually provided with either an automatic power factor regulator or an automatic voltage regulator. They materially improve line regulation and reduce power costs. Heretofore, the large power supply corporations have been largely instrumental in the placing of automatic synchronous condensers at strategic points on their power system. It is believed, however, that steel mill engineers will soon recognize the advantages to be derived from the placing of automatic synchronous condensers at various points on their distribution systems and that the advent of the automatic will permit them to make wider use of this very important corrective feature.

Synchronous converters and synchronous motor generators of the smaller sizes equipped with automatic features are now being rapidly installed by all coal and metal mining operators. Their design resembles that of the synchronous motor generator equipment previously described excepting that on account of their smaller size, no attempt is made to provide inherent automatic load regulation. Sometimes, the feeders of such equipments are made load limiting, but more usually they are made automatic tripping and automatic reclosing. In case of an overload or short circuit on the system to which one of these feeders is connected, the circuit breaker automatically opens. This shunts a high resistance into the feeder circuit and thus permits a small feeling-out current to flow. Suitable relays in combinations with the feeling-out of load indicating resistor determine when the short circuit or overload has been removed, and then the feeder is automatically reconnected to the bus. This type of reclosing feeder is well adapted for certain classes of steel mill service. For other classes, the discriminating type of feeder is recommended. This design is so arranged that the feeder circuit is only opened on short circuit or extremely long continued overload. When opening, it does so with such promptitude that the circuit current cannot rise to a sufficient value to cause flash-over of the d.c. machines. The feeder remains disconnected from the bus as long as the short circuit or other abnormal condition exists. When the feeder becomes normal again, it is automatically reconnected to the bus. Sometimes such a feeder is provided with a continuous feeling-out arrangement, sometimes with intermittent feeling-out. The type of feeling-out design depends to a great extent on the service conditions.

Automatic switching equipments in general protect against practically all hazards which may be encountered in the operation of electrical equipments. A brief review of the features incorporated in their design shows that they:

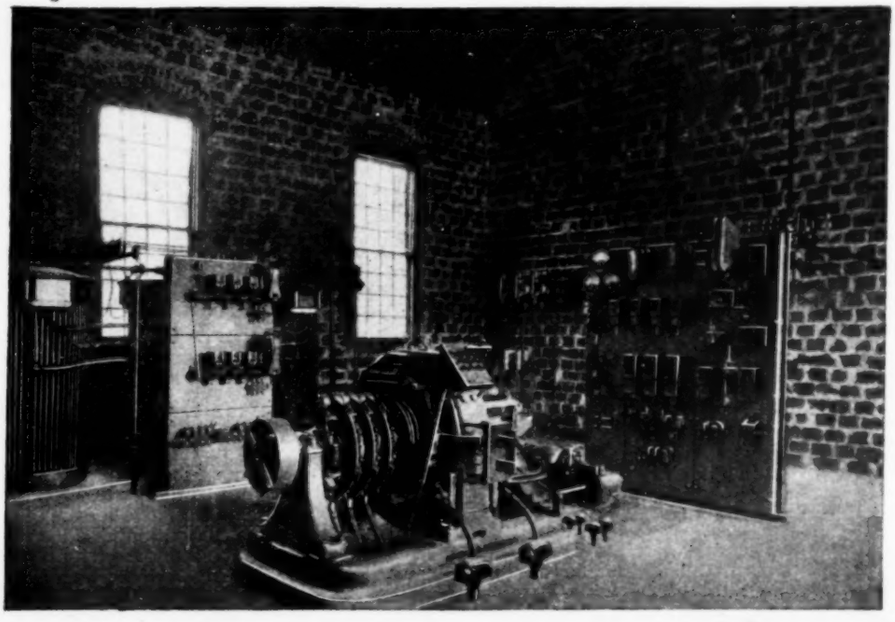

FIG. 6.

1. Start the equipment when the indications are given by the master starting element if the correct conditions exists.

2. Prevent starting the equipment if:

(a) The a.c. line voltage is below a safe operating value.

(b) The a.c. power supply is single phase.

(c) The set has been shut down as described under (5) and (6).

3. Connect the equipment to its load within a short time after the starting sequence has been completed or it has been disconnected due to overload or reverse power and normal conditions have returned; if:

(a) The polarity is correct.

(b) Generator voltage has built up.

(c) The motor is connected to full line voltage.

(d) The motor field is excited.

(e) The load is not excessive.

(f) Voltage of machine is higher than that of bus.

4. Shut down the set when the stopping indication is given by the master starting element.

5. Shut down the set temporarily if:

(a) The voltage of the a.c. power supply is too low for safe operation of the equipment.

(b) The set is subjected to moderate continued overload which, if continued longer, would overheat the set.

(c) The set is subjected to single or badly unbalanced phase operation sufficient, if continued, to cause overheating of the machine. The set can restart when the windings have cooled or voltage has been restored.

6. Shut down and "lock-out" the set if:

(a) The equipment is subjected to harmful a.c. overload sufficient to require inspection adjustment or repair before being placed in service again.

(b) The field of the synchronous motor should fail.

(c) The set should tend to overspeed.

(d) A bearing should tend to overheat.

(e) The motor is not connected to the full line voltage within a definite time after the beginning of the starting sequence.

7. Disconnect the generator from its load if:

(a) d.c. overload occurs.

(b) Reverse current equal to or greater than the normal running light current of the set occurs.

(c) The d.c. voltage fails.

(d) The set is shut down.

This review indicates briefly the operation of a typical automatic station and the application of some of the equipments. Besides it mentions several unique designs which are especially adapted for steel mill service.

Discussion on Ball and Roller Bearings

D. M. Petty*: The work which I have done in connection with ball bearings has been primarily to do a lot of talking and get the other fellows to do the work. We have tried a few installations at Bethlehem and are trying others. This is the first of a series of meetings before sections on the subject and the object of the committee in asking the various sections to hold these meetings has been primarily to assemble from various papers the different combinations of ideas for mounting and maintenance. Those users of motors, of course, who will contribute to the discussion will tell us more about maintenance than the fellows who are designing the bearings. It seems to me that after we have completed this we all ought to be able to get some idea of how we can attack this problem of mounting roller or ball bearings in motors. I feel that the two speakers here tonight have given us just about what we want. I was very glad that the ball bearing man gave us some of the theory of design and building practice and of what a ball bearing actually does. I have never been particularly interested in the design, but I have been interested in securing a bearing in a motor that would work, but I do believe that we can all make more intelligent use of the bearings if we understand a little bit as to some of the limitations and the factors which the designers have to overcome. I had an interesting experience the other day in asking my foreman, in our regular weekly meeting, if they had made it a practice to lubricate the various ball bearing motors which they have under their supervision. I was not surprised to get the answer from them that they did not know, and one of the foremen was frank enough to say that his had not been lubricated at all. We started out after the meeting with the definite idea of finding out what was the best time or the maximum time in which you can leave the ball bearings run without lubrication. We have all kinds of freak cases on record where ball bearing motors have run for years without any additional lubrication than that which was originally packed in with it. In talking on the subject of how often they do need to be lubricated, Mr. Cummins, of the Duquesne Plant of the Carnegie Steel Company, was telling me recently of his experience. They had previously adopted the practice of lubricating their roller bearing motors once a month, but after hearing the various discussions for the past year or so on the subject they decided that they could stretch that over two months. They found that by stretching it to two months, in a few cases of exceptionally hard service, and in cases where the temperature is rather high, they got into a little trouble; so, as far as I know now, they have gone back to the idea of lubricating those motors once a month. They are using grease entirely. The comments on that particular problem were that no grease apparently ever came out of the bearing, that a little bit was put in every month, but what became of it is still a mystery.

I think after we are all through with these various meetings and the papers and discussions are published that we will all have a little more definite idea as to what we can and cannot do. As far as standardization is concerned, whether we should adopt the 300 series or some other series, I do not know. We had this pretty well threshed out at the Buffalo convention, but some additional factors have come up since that time in the Power Club which will be more or less ironed out next week, so that it seems altogether possible that what is considered standard will be announced about the same time in which the results of the meetings this month are published. I believe that some time in the near future we can expect to have a fairly good resume of the situation as it stands now.

I do not feel that I can add anything to the discussion of the papers that have been presented tonight.

*Superintendent Electrical Department, Bethlehem Steel Company, Bethlehem, Pa.

I feel that we have done quite a good deal of talking about ball and roller bearings, and we have not heard to any great extent from the designers and builders of these bearings, which seems to me to emphasize the point that we should let the bearing engineers tell their story this year before we go too far with the subject. Of course, I realize that none of us would hardly presume to go ahead with installing bearings in our motors without consulting rather freely the designers and the builders, although I do think, as has been pointed out here tonight, the users of motors have been more willing to take a chance than the builders of the bearings. I feel if there is anyone here that has had any experience, particularly those experiences in which they have found trouble, and can point out the things that should not be done in installing bearings, it would be well to get them into the record at this time, because some of those things will be just as useful as knowing those things that can be done.

D. E. Batesole*: The papers were so very well presented and so much valuable information given on the subject that there is very little to add. In particular, we believe the slides showing the various mounting details of roller bearings as presented by Mr. Wachtler, were very instructive. In connection with the mounting of Hoffmann roller bearings we believe the use of a deep groove ball location bearing to be very desirable in connection with them. This bearing is used of course to provide endwise location to the armature shaft, thereby doing away with the friction caused by the rubbing of one friction surface against another. Even though these rubbing surfaces be well lubricated, there may be considerable friction if the end pressure is great due to an endwise magnetic pull of the armature shaft. We have used this ball location bearing sometimes alone at the closed end of the motor to take care of radial load as well as end thrust and sometimes with a roller bearing at this end. In the latter case the roller bearing carries the radial load and the ball bearing the end thrust load. On the drive end of the motorshaft where the load is very heavy, the roller bearing is of course recommended.

F. W. Gurney†: I have, of course, been interested in what Mr. MacCutcheon has just said, and also in what Mr. Crow said about ball bearings. As to the theoretical part of Mr. Crow's discussion about the design and construction and the general principles of ball bearings, with a great deal of it I am in agreement, and in some of it I would have to differ a little bit. The points he claimed or made with regard to there being no slippage in ball bearings was a little bit extreme. He spoke of the compression of the ball giving a barrel shape to the ball. He overlooked the point that there is also a compression of the raceway, the race-way giving just as much as the ball. There is actually a considerable difference in diameter in the portions of the balls that are in contact with the raceway. If the ball is rolling in its inner portion it must be slipping in its outer portion, or if it rolls on the outer portion of its tread it is slipping in its inner portion. Of course, the fact of the matter is, there must be a compromise between the two conditions; it is slipping a little both on the inner and outer, but there is a neutral position somewhere between where there is no slippage. But the slippage in both cases is so slight as to be practically negligible.

There is another source of friction or of heating which Mr. Crow probably would have referred to if he had thought of it, and that is you get what we call the molecular friction. You know if you take a little bar of iron and bend it a few times it gets hot. The same thing exactly is occuring in the steel of the ball end of the race-way. I think there is, contrary to what Mr. Crow said, a back and forth movement of the steel in both the balls and the race-way, that is, a little waving of steel proceeding just ahead of the ball. Where the steel is actually coming up and going down this waving, this to and fro movement of molecules of the steel, causes heat, a great deal of heat if the load is very severe.

There is another point; the angular contact or the ratio-thrust bearing has a very decided spinning action, and the ball, even if it is rolling on the same axis still spins and the angular contact bearing the ball is running on what may be called a cone, or rather a short frustum of a cone. The apex of that cone is at the point where the tangent of the ball with the race-way crosses the center of the shaft. That ball is actually rolling on such a cone. If you would unroll that cone you would find that your ball is rolling around a circle whose radius is the tangent from that ball to the center of the shaft. The balls roll around a circle, and so is doing two things, it is rolling, it is also spinning. When it rolls around a circle it describes 360 deg. of spin. For that reason the angular contact bearings run warmer than a plain radial bearing. Under this spinning the two surfaces between which the ball is rolling are turning with reference to each other like that (indicating), so that the ball is of physical necessity spinning between those two spinning surfaces. It is the contacting element between two surfaces spinning with reference to each other. That is the particular source of the friction and the heat of an angular contact or radio-thrust bearing. The friction, you may say, under constant loads and speeds of ball bearings is in inverse proportion to the tangent of the angle of inclination. You may consider all bearings angular contact bearings. The ordinary groved thrust bearing is the bearing whose angle of contact is the greatest, 90 deg., and the radius whose circle is the shortest, and whose friction is the greatest. You cannot run that type of bearing at very high speed under large load. At very high speed the angular contact bearing is much more efficient.

I was much interested in what Mr. MacCutcheon said about standardization of motor bearings. This is something that must come ultimately, but it cannot be forced. Mr. Crow and some of the other manufacturers are very familiar with the work of standardization of ball bearings in the S. A. E. We know perfectly well that we cannot arbitrarily standardize bearings; we can merely recognize these standards as they take place; standardization comes as the result of experience; it cannot be forced upon the industry by anyone manufacturer of motors or manufacturers of ball bearings either.

F. W. Caldwell*: One of the curves which was shown tonight as representing the friction characteristics of bearings unfortunately was taken so that

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*General Electric Company, Schenectady, New York.

it does not represent what happens at speeds at which the ordinary motors run. Bearing speeds of about 100 to 150 feet per minute are seldom used by motor manufacturers, the minimum being usually about double these figures. A number of years ago in an attempt to find out the difference between the friction of a roller bearing and a sleeve bearing, a 25 hp. induction motor fitted first with a sleeve bearing had a flywheel made from a solid steel disc 33-in. in diameter with a 6-in. face, weighing 1400 lbs. mounted in place of a pulley. The motor was speeded up to 1800 revolutions per minute and the power cut off, careful readings being taken of speed-time curve during deceleration. After several repetitions, the bearings were changed and roller bearings were substituted using the same armature and flywheel in order to keep all of the losses except the bearing losses as nearly constant as possible. Out of several runs, there was considerable variation in the results with each type of bearing but the average result was somewhat surprising. In about 20 minutes the motor and flywheel would drop from 1800 to 600 r.p.m. with either bearing. The greatest difference in time was about ¾ of a minute. On the other hand, to drop from 600 r.p.m. to standstill took 38 minutes with a roller bearing and 26 minutes with a sleeve bearing. When it is considered that 25 hp. motors are seldom run at speeds below 600 r.p.m., it is evident that in this case there was practically no difference between the friction of the two kinds of bearings at motor speeds but that from 600 r.p.m. down which are speeds below that at which such a motor is ordinarily run that the flywheel on the roller bearing ran nearly 50 per cent longer than on the sleeve bearing. The bearing load in both cases was a considerable overload or 125 lbs. per square inch on the sleeve bearing.

There are two general methods for applying ball or roller bearings to motors. The first is for the slower speed apparatus where the capacity of the bearing ought to be close to the capacity of the shaft under repeated stresses when applied throughout its life. The second is for higher speed apparatus where a bearing is selected which will properly function under the conditions of the application without causing too much heating. It has been possible to work out very satisfactory results in connection with sleeve bearings to cover these cases of application but there still seem to be certain factors which require considerable engineering attention in the case of the ball and roller bearing applications in both classes of service. Under the first class, if a shaft is run continuously under repeated stresses to as high a limit as it can stand for a long period, the application of a ball or roller bearing to this case might possibly be analyzed along the same line as the shafts. The ball or roller turns a different number of times than a shaft does and has a considerably different elastic limit. By properly proportioning the elastic limit and load and the number of revolutions to be made by both in their life time, suitable average factors could be worked out so that one would be able to predict under equal conditions what would be the best arrangement.

The discussion of the stresses in the clamping nut reflects on the fitting of the inner race of the bearing to the shaft. The system of stresses which takes place in the combination of a slightly eccentric improperly grooved raceway with a tight fit on the shaft in combination with a clamping nut unable to support a great amount of load is very complicated. The problem of the clamping nut is much simplified if the raceways can be sufficiently long and heavy so that there can be more leeway in the fit. It is found in general that with long hubs and a moderate amount of bending, the size to size fits are quite satisfactory. As the hubs are made shorter and bending increased, pressed fits with considerable pressure are necessary but if we go still shorter so that the length of hub is less than one diameter, it is necessary to have an extremely heavy hub and shrink it on almost red hot in order to have the parts stay on under a bending stress. The amount of pressure between the two members might be investigated with the photo-elastic method using polarized light and transparent models. Some very interesting results have been brought out in the case of pinions shrunk on the shaft and subjected to gear tooth loads both when standing and running. An interesting feature that has been brought out is that the stresses which go through the hub to the point where it bears on the shaft are of considerable magnitude. I have felt for a good many years that by widening and thickening the inner races of ball and roller bearings, more rapid progress in application might be made. The percentage of bearing failures due to the loosening of the inner race on the shaft are of such an order that failures from this cause require careful scanning.

If the inner race can be widened, less attention will probably be needed to the clamping nut although attention might be drawn to a construction used on certain light weight railway motors where a shrink ring is pushed on outside of the bearing race-way entirely replacing the nut. In the case of this type of motor, the shaft stresses run much higher than they do in the ordinary industrial motor and this method has been found to meet conditions better than the nut although it requires some special apparatus for removal in the case of changing a bearing.

A question which I wanted to ask was in regard to the 200 lbs. per inch width of belt used in calculating the load on a motor bearing. Was it intended that there should be 200 lbs. on the tight side and 200 lbs. on the slack side of the belt or 200 lbs. on both sides together? Some of the calculations which I have recently seen appear to indicate that the two are taken as one. It may be that Mr. Wachtler can straighten out this point for us.

Arthur H. Allen*: There is one question I would like to ask with regard to the shock on ball bearings in the operation of motors. It would appeal to me, as far as my personal experience is concerned, that a motor operating under shock conditions, is not an ideal condition for a ball bearing motor. Personally, I would very much prefer, of the two types, the roller bearings. I notice that the gentleman who spoke in regard to ball bearings, omitted that phase of the practical operation of bearings; while the roller bearing man made a point of the shock condition which is very frequently experienced in the operation of all motors, either belt driven or gear driven, and it is a practical operating condition which I think many ball bearing advocates have not taken consideration of. I would like some expression on this point.

F. O. Schnure†: The application of anti-friction bearings to mill type motors calls for consideration of

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the size and type of bearing and its adaptability to the present sleeve type bearing housings. The size of bearing to be used on mill motors should be determined by the most conservative design, allowing the greatest possible leeway in favor of a rugged bearing in order to withstand shock loads and the extraordinary conditions that arise in mill service. I believe that in the adaptation of anti-friction bearings to mill motors we will find some difficulty in maintaining a fit between the inner race and shaft, and therefore agree that the best fit will be secured by having length rather than thickness of the inner race. As all of our mill type motors now have sleeve type bearings with mechanical clearances accordingly, there is plenty of room in designing an anti-friction bearing motor to allow for a long bearing. We are of course more interested in being able to adapt the anti-friction bearing to the present motor housings rather than in their replacement with new housings or entirely new motors which procedure would require considerable capital expenditure. New housings for mill motors represent about 40 per cent of the cost of a new motor. Anti-friction bearings, and alternations to the armature shaft should not amount to over 20 per cent of the cost of a new motor; therefore, any arrangement that will utilize the present housings represents considerable saving. With general purpose motors the point is not so serious, as end bells are not very expensive.

With regard to the lubrication of anti-friction bearings it seems to me that grease can be more easily retained in the bearings and would therefore not get out to cause commutator trouble, the serious motor ailment we hope to cure.

Another objection I find with oil for lubrication lies in the experience we have had on the operation of ball bearing headlight turbo-generators. We started out by using a light engine oil and found that considerable trouble was experienced, due to rusting or tarnishing of the bearings, at such times as the locomotive was out of service. Twenty-four hour periods gave us quite a bit of trouble and of course longer periods of disuse meant greater bearing deterioration. The trouble was aggravated by the salt atmosphere present in our plant, and was partially cured by the use of a light grease or heavy cylinder oil. It seems to me then that mill motors which are subject to weekly, monthly or even yearly shutdowns would show considerable bearing deterioration unless a heavier oil or grease was used that would not entirely run off the polished surfaces.

The lubrication of the thrust collar as shown in the drawings presented by the author of the roller bearing paper, looks like a problem that might give us more trouble than the lubrication of our present sleeve bearings. Some of our drives may line up all right when installed, but at times, mechanical troubles may develop thrust in one direction or another, so that unless this collar was so designed that it would take this thrust to the maximum amount without injury, the bearing would not serve the mill satisfactorily.

The author of the roller bearing paper's presentation of his method of changing an armature, relieves us of the fear that during this process we would get dirt or foreign matter into the anti-friction bearings. This has been one of the anticipated objections to this bearing, as many of our armature changes are made under rather adverse conditions by semi-skilled help. Experience has shown us that it is very difficult to keep dirt out of the sleeve bearing during this process. The anti-friction bearing will come into more general use in the Steel Plant, just as soon as a type of bearing is developed that can be installed in our present housings at very little expense.

H. A. Lewis*: I have listened to the paper with much interest, but owing to the lateness of the hour, I was unable to wait and make a few remarks that I had in mind.

I do not see the reason for changing from a perfectly good bearing to one that has not had the severe test of a steel mill. We have been using ball bearings on crane hooks as well as on some revolving overhead billet charging cranes and as yet we have not had success with ball bearings.

A few years ago the automobile manufacturers placed an engine with a ball bearing crankshaft on the market, but it quietly disappeared.

The most experience we have in ball bearings is on our automobiles. Have you ever compared the mileage of an automobile to an electric motor? The mileage on my own automobile for one year is 10,000 miles. An electric motor going at half the speed, say 750 r.p.m., would make the same mileage in 28 days. After an automobile has run 10,000 miles, take it apart for an examination and you will find at least some of the ball bearings damaged. You have babbitt and bronze bearings on the crankshafts and wrist pins traveling at a high speed. Upon examination you will find them in good condition and some of the balls or rollers damaged, by doing less work, on the same machine.

At this point I wish to quote the experience of the largest and best managed corporation in the world. This speech was delivered before the Engineers Society of Western Pennsylvania by Mr. Joseph Slutzker:

"I am with the Pennsylvania Railroad and we consider a good daily operation about 8,000 loaded cars eastbound and approximately the same number westbound. There is about 135,000 bearings running across our division daily. In our passenger service we have probably 50,000 brasses running across our division. In the passenger service we tabulated the number of mechanical failures. Again with that in view, a series of tests was run by the Pennsylvania Railroad with ball bearing journals to see how they would compare with brass bearings. The starting resistance of the train with ball bearings was very much less than the starting resistance of a train of equal weight with brass bearings, but the resistance of the train after getting to the speed of 45 to 50 miles per hour was approximately the same. We have many trains that start at Pittsburgh and make three stops between Pittsburgh and New York. So in view of that fact and the very powerful locomotives that we have, the starting resistance does not amount to much. The ball bearings were run a considerable distance, probably covering almost two years and much difficulty was encountered with the balls. It was decided therefor to give up the ball bearing test and go back to brasses."

Just before coming down to this meeting, I requested my office force to give me a report on the

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number of armatures in the shop for repairs during 1923, and I found as follows: Thirty per cent of the total amount of armatures in use were in the shop; 27 per cent of this number were rewound, 2½ per cent had defective bearings. The experience we have is that a new motor can be used for quite some time without causing any trouble, but as soon as it is repaired the trouble begins. Now if we cannot maintain the present motors with our organization, I feel sure it would be unwise to encourage or select a piece of machinery costing considerable more money, requiring a higher grade mechanics to apply and maintain the same.

T. C. D. Crow†: Regarding the use of headless set screws in split locknuts, will admit that this type of screw is not so easy to tighten as a hexagon head screw; therefore, the latter could be used to advantage providing sufficient clearance is available between the nut and bearing housing.

Regarding the tapping of the hole in the split nut, you either have to tap in the thin or thick section. There is considerable strength obtained and also a mechanical advantage gained in tapping the thick section of the nut.

Regarding expansion of the race, a bearing can be assembled with sufficient radial clearance to compensate for mounting expansion, but the variation in mounting conditions makes it necessary to select bearings to suit the fits used. One builder of gasoline motors is using .0015 press — the bearing is very loose before assembly but after mounting on the shaft there is practically no radial clearance or end-play.

It is hard to tell when a ball bearing is assembled too tight as it is possible to assemble a bearing in which the diameter of the inner race plus twice the ball diameter is two or three tenths larger than the outer ball race and yet have the bearing spin freely. You may, therefore, have an initial load created greater than the pay load if the bearing is assembled too tight.

It is possible to mount the inner race on shaft without locknuts. There are two reasons for locking —one is to prevent endwise motion, the other to increase the holding power of bearing on shaft. However, as long as the bore of the bearing is smaller than the diameter of the shaft, there can be no motion. If, however, the bore of the bearing is larger than the shaft, you have a very high reduction in internal friction drive and no locknut will prevent rotation in this case. There are motors built at the present time in which the bearings are not locked on the shaft, the endwise motion of the armature being controlled by allowing the overall dimensions between the shoulders of your end-bells or housings to be slightly greater than the overall dimensions of the bearings.

Regarding Mr. Gurney's remarks as to spinning of balls, I am afraid Mr. Gurney misunderstood the particular points brought out and a technical discussion between Mr. Gurney and myself would be lengthy and not of interest at this time.

The remarks regarding that in a correctly designed ball bearing all parts would fail at the same time, the next approach to the "one-horse-chaise" would be a design in which all parts were of equal strength. Our own endurance tests show that failure of the inner and outer races and balls are equally divided, indicating that all parts have the same endurance qualities.

Regarding shock load, this is a much debated point. The only way that shock can affect a ball bearing is the tendency to split the ball and the ball can only be split by a blow directly over its center and this would be hard to do when the bearing is rotating. We have a shock testing machine in which we apply a test load of twice the bearing's rated capacity at a given speed. Added to this load is a weight which is dropped like a trip-hammer. However, the machine breaks down more often than the bearings; hence, the results are rather inaccurate. Would not quote any definite figures, but would say that the reduction in testing life of the bearing is very slight and can readily be taken care of by a slight increase in factor of safety.

Regarding factor of safety recommended, this is a variable depending on the type of machine, which is good engineering. The factor, which is the capacity of the bearing at a given speed divided by the load, will vary from one to three. It is interesting to note that tests so far have shown that if a bearing is not overloaded, the life obtained in continual running is no greater than intermittent running.

Regarding the statements on standardization, I can see no reason that for general purpose motors it would not be possible to select the bearings for a given horsepower at a given speed rating, based on an average spacing between bearings and pulley overhang, though I do feel that the same bearing can be chosen to meet the high speed and low speed conditions of a given frame size. As a matter of fact, we have a table showing bearings that we are perfectly satisfied to recommend for general purpose motors at the most popular speeds which experience has shown will give satisfactory service and therefore, Mr. MacCutheon's remarks are not radical by any means as far as we are concerned.

R. Pruger*: I am sure that we all enjoyed the fine talks here this evening, and that we all gained some valuable points. I particularly appreciate the fact that the ball and roller bearing engineers are making strenuous efforts to take up the problems of the motor manufacturer. Some few years ago they were not nearly as much interested in our problems as they are now. This was proven by the papers presented here this evening, and which showed a close study of application of anti-friction bearings to electric motors.

A number of experiments carried on by ourselves, and by other motor manufacturers as well, have shown a number of facts which coincide with what we have heard here this evening, particularly as regards the mounting of ball and roller bearings, as pointed out by Mr. Buckwalter. It has been my experience that correct mounting is a very imporatnt feature when using anti-friction bearings, and that failure is oftentimes due to errors made in this regard. It is only a few days ago that I returned from a trip, having visited a plant where ball bearings were used on gear units. I may be free to state that the bearings were of the Gurney type, and also, that they were good bearings. Unfortunately they had been mounted with too large a press fit allowance of the inner race on the shaft, which, of course, expanded the inner race to

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such an extent as to cause a binding, thus creating considerable friction and wear. In addition to this there was wear of the bronze worm wheel, and particles of bronze entered the bearing, there being no provision made to protect the bearing. Heat generated in this gear unit caused carbonization of the lubricant, and the sticky compound of carbon residue and bronze particles were allowed to enter the bearing freely; this, in combination with the tightness of the bearing, caused a break-down in a short time. Checking up bearing reactions, speed and load, it was found that the factor of safety was ample.

Another item of interest in connection with bearing mounting is the exactness required in machining. We are probably all agreed upon that the outer race of a correctly mounted ball bearing should be free enough to rotate or "creep"; any slight bore variation in housing may easily prevent this, causing the outer race to stick. In that case, wear is concentrated upon a short arc of the outer race, and may thus cause failure in a short time. Such a case came to my attention only very recently and when checking the bore it was found that same was .001 in. out of round. It may thus be seen that in actual field cases, a close investigation should be made before dismissing the matter as a case of "defective bearing". I do not state this for the purpose of taking up the defense of the ball bearing manufacturer, but merely wish to point out, that those who are using ball bearings should carefully observe the rules for mounting as given by the ball bearing manufacturer; otherwise, trouble may easily result, the blame being ours for neglecting instructions.

It is obvious that the field of anti-friction bearings is quite a large one, and the tendency to make use of such bearings is increasing continuously, even though the problems of the motor manufacturer as well as other ball bearing users are quite complicated. It may be fairly simple to decide upon a certain type and correct sizes of bearings for a line of motors from 5 to 50 or 75 hp. for ordinary belted application; but if you consider the large variety of applications and severe service conditions, you may probably agree that selection of the correct sizes and type is not quite such an easy matter. To illustrate this point, I may refer to an actual case where ball bearings were in use on a number of machines operating under high loads and slow speeds, starting and stopping frequently. It was thought that the ball bearing was the proper thing to use, since it eliminated the high starting friction common to sleeve bearings; however, facts proved that the ball bearings did not stand up, even though of liberal proportions. This led to the conclusion that for this particular application roller bearings were the proper thing to use, and which fact was borne out by experience later on. On the other hand, in another case, roller bearings were tried out on high speed—3600 r.p.m., and which was in the nature of an experiment. Although there was practically no load on this bearing, it failed in a short time, the cage breaking, and causing destruction of the entire bearing. Replacing this roller bearing by a ball bearing, no further trouble was experienced.

Another very important feature, and which was mentioned by both of the able speakers of this evening, is the matter of lubrication. I shall merely mention an incident where grease was used, and where considerable heating was noticed, the machine operating at high speed. This temperature rise was not due to the fact that grease was applied, but due to the quantity in which it was used, the bearing having been packed full. Removing about 2/3 of the grease resulted in considerable temperature drop. This same point was brought out at a recent meeting here in Cleveland by one of the ball bearing engineers, and it coincided entirely with the experience which we had ourselves.

As regards the matter of excluding dirt, I may say, that felt washers are commonly used for that purpose. However, if compressed close to the shaft, the felt will soon glaze and burn out, permitting oil or grease to leak out, and dirt to get into the bearing. It was found that soft felt washers should be used, but held in place and compressed by a sheet steel washer which has an inside diameter of at least ¼ in. larger than the shaft; this permits the felt to expand where it touches the shaft, and retain its resilience, acting more in the nature of a brush or wiper without hardly any wear, and effectively excluding dirt. This little wrinkle has been used with good results, and in view of the fact that it is of great importance to keep anti-friction bearings free from dirt and dust, it is to be recommended.

E. W. P. Smith*: Our experience so far as grease has been concerned in lubrication has been very similar. We have had difficulty with the 310 running at 1800 r.p.m., carrying a rotor weighing 300 pounds, and with too much grease in the bearing. We took out about ¾ of the grease and it ran fine.

I was very much interested in the remarks of the Westinghouse representative when he said to look for the difficulty in the installation of the ball bearings. We have had some cases where the ball bearings have failed and we have really no explanation of the failure. We are trying out some other schemes to see if we can find out what the cause is. Perhaps that is the reason or the answer. So far we haven't been able to answer that because apparently the installations were in good shape.

I'd like to know just what speed some of the operators find the best at which to change from grease to oil, or do they object to oil as a method of lubricating ball bearings; and if they do favor oil do they prefer a scheme of what might be called forced lubrication in which the rotating of the shaft forces the oil into the bearings or something in which the bearing runs in oil? Both, I believe, have distinct advantages and both have, of course, disadvanatges.

Philip Chapin Jones†: I have watched the advent of the anti-friction bearing for some years and I have felt and still feel that it is absolutely bound to come. It is one of those things that we cannot escape. There has been opposition from many manufacturers, largely a disinclination to make the necessary changes in their designs to allow the use of the anti-friction bearing, but such opposition is no more than a delay. The only thing to be decided is what types will be used or where we shall use one type and where another.

I think the anti-friction bearing is bound to come for three or four reasons. The least of these reasons is that there is a lower friction loss and thus a higher efficiency. One very strong reason is that a small motor equipped with anti-friction bearings may be put

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in inaccessible places where it will not receive attention and still give service. Then, too, the fact that bearings of this type can be made tight, thus keeping all the oil out of the winding, is a very great advantage.

There probably is a place for the sleeve bearing as well as for the roller and the ball bearing. I think the principal problem is to determine those places.

I should like to hear from the roller bearing people as to what success they have had with the higher speeds. From what I understand they become noisy at the higher speeds and do not give satisfactory operation. Probably in the very large motors and in the lower speed motors we can use the roller bearing and the sleeve bearing and in the higher speeds and smaller sizes the ball bearing.

E. E. Eby‡: I thought I was going to dodge this tonight because Mr. Wachtler gave our talk down in Philadelphia, but I might assure this, that we are interested in a discussion on mountings principally and some of the problems that you men are up against in the maintenance of motors. I'd like to see some of you operating men cross-examine the bearing people as to what they propose to do under certain conditions. I think that is a problem that confronts the bearing people today. We have bearings running under very severe loads in motors in every industry, principally the steel industry, and the problem that confronts us, and I think the ball bearing people as well, is a suitable mounting to take care of your operating conditions. When I say "operating" I mean maintenance, tearing down and all of the problems that confront you, because I don't believe you will have the skilled men in many instances in the mills that you do have in the tool rooms, etc.

J. Kelly§: The bearing people and others speak of high speeds, low speeds, and heavy loads and light loads. What do they mean by high speeds and light loads, and low speeds and heavy loads? Some of us may think that 1800 r.p.m. is a very high speed; others may think it is low speed. The motors in mill service probably never get up to 1800 r.p.m., while other equipment in plants such as saws and things like that operate at those speeds. Various types of anti-friction bearings have been tried out. Some have proved satisfactory and some have not.

The steel plant operating men are not electrical engineers. Generally, they are connected with the mechanical department, and they can tell you more, I believe, about the speeds and the loads and shock loads which are put on this equipment than any other steel plant equipment. Reversing motors, no doubt, get a decided shock, and any bearing put in such a motor will have to stand them. Whether it should be ball or roller, I think many electrical engineers can find out by talking to some of the master mechanics and maintenance men in these mills.

Samuel Farrell*: I am glad to agree with the remarks of Mr. Kelly of the Hyatt Company. I think that most of the high speed motors are usually installed with pumps or saws or something of that sort, and usually have a building of their own, and I think they get more attention and more inspection than the average mill type motor that is used, because the average mill motors are in operation all the time and the inspector hasn't an opportunity to work on them except now and then.

I think the question of bearings is being brought about by motor troubles, and I think the sleeve bearing due to the numerous reversals of an armature has caused a spray of oil to arise from what we might call the face of the bearing and shoulder on the shaft, and with the rotor permanently tied in its field, the same as that sketch shown there, you would not have any occasion for oil spray and with the bearing sealed you would get better operating conditions.

A. J. Porskievies†: There are several questions I'd like to ask these gentlemen that presented these papers. One of them is the mounting of the inner race on the shaft. To my mind that represents the most important point in the mounting of ball or roller bearings. If the inner race is not mounted properly, as these gentlemen brought out, the ball or roller bearing generally goes to destruction. If, on the other hand, you inquire and ask these ball bearing manufacturers or roller bearing manufacturers, what press fits they require for a given shaft or a given bearing, you will find them varying all over the lot. That is, by that I mean you almost have to specify the duty under which the motor has to operate. On industrial motors that is almost utterly impossible to do. That is one point I'd like them to clearify for us or give us some explanation on.

The other point is on a motor that is designed properly, with its proper installation on the windings, the amount of oil that it may throw on the windings is not disastrous. And then again, if the bearing housing is designed properly there is no need of the bearing to throw oil.

Wm. T. Burns*: One thing I might say a word on is the lubrication. Mr. Gurney spoke of the various causes of friction. The character and the amount of lubricant has a further important bearing on the subject. We have run many tests using various lubricants, beginning with kerosene and going to heavy grease, and under the same load conditions the bearing temperature is very closely proportional to the viscosity of the lubricant, so that by using a lubricant of sufficient body to be held by the enclosure at the speed the motor is running, and still one not too heavy to run up the temperature which is due to the actual mechanical work that is done on the lubricant, we can get a satisfactory job. I don't think I can touch on any of the other subjects as Mr. Buckwalter has no doubt a line of answers that will take care of them.

D. E. Batesole†: In connection with this question of speed as applied to roller bearings I might say that while ball bearings are a very well standardized product, in other words, made up with practically the same steel specifications and tolerances in manufacture, this same thing cannot be said of roller bearings at the present time, so in considering this question much depends upon the type and construction of the roller bearing. We have had extensive experience with the Hoffman roller bearing which is made up to

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†The Norma Company of America.

very close tolerances, running at speeds of 3,600 r.p.m. on electric motors. We also have been running them at 7200 r.p.m. but this application was on wood-working machines.

We have found from our experience that roller bearings of the light series are better suited for very high speed work. I recall one case where a manufacturer had been using a heavy series ball bearing and put in a heavy series roller bearing of the same size. At the high speed it was found that too much heat was generated. However, by going to the light series roller bearing with the same shaft size he found that no more heat was generated than with the ball bearing previously used.

It is our opinion that the heat generated by the heavy series roller bearing was due to the large rotating mass at the high speed which naturally caused considerable friction.

H. C. Cronk‡: Referring to the question of proper lubricant and properly protecting the bearing, it seems to me that we would get very much better results from sleeve bearings if we took as much pains protecting them against leaking out of the grease and infiltration of air as we do with roller and ball bearings. We don't pay any attention to that. We simply put them in within sixteenth of an inch fit, pay no attention to the protection against dirt and retaining the lubricant, and then we expect to get a long run. When we come along and put in a roller or ball bearing, we have the finest mechanic in the place to fit it up, we work down to thousandths of an inch, and we are directed by the roller and ball bearing manufacturers to put on all kinds of protection with very good results.

That may sound as though I am not an advocate of the roller or ball bearing. I can go along and illustrate a few things that will show you they are very worthy. We had an illustration of a five hp. motor driving a fan at a metal mixer. Any one familiar with mixers understands the dirty conditions existing there. The bearings on the one end of the motor, belt drive, would run one week. The bearings on the two ends were duplicates. When we brought the motors in, which we did once a week, we changed the bearings. We would take the bearing out of the end opposite the pulley and put it on the pulley end, by this method we would get another week's run out of it, making two weeks maximum.

We went out to the scrap pile—tons of it laying around the plant, and found some old automobile axles. Somehow or other the scrap dealers didn't get the bearings. We took them out, put a set in this motor and they worked fine. The first run we got out of it was nine weeks when the winding broke down. We just let it run. The limit then became the winding.

Now you gentlemen who say that oil and grease and dirt on our windings don't do it any good, are certainly correct, but when you come to referring to the mill duty, it is our worst enemy. The oil begins to accumulate and other dirt and dust, graphite in the various locations, and the result is a breakdown.

Another case where we used the same method of obtaining bearings was on a 10 hp. motor, running 750 r.p.m. It was on a conveyor at our coke plant. The ball bearings were installed and the motor put in service. When we got it in place it ran very hot. It was not a case of too much grease either. We looked it over and decided to line it up properly. That motor ran 17 months. Our best run usually was 7 months on sleeve bearings in this particular motor. We didn't take the motor out for any failure of the bearing. We took it out because we were beginning to get uneasy about the windings. We put a sleeve bearing motor in its place and ran it a few months and then wanted to continue the run on this motor, put it back in and it was in service up until a few days ago, when the load became so great at this particular place we had to put in a larger size. Had there not been this roller bearing equipped motor at this location, we would have had to change that motor and increased the size several months ago.

These are a few of the operating man's experience.

In all our installations on anti-friction bearings, we have been putting in both ball and roller bearings, we use grease entirely — don't pretend to use oil. We can hold the greases easier and it is much better to handle and less costly, of course. We think it is a great deal better than the liquid lubricant.

I said in opening that we do not find it difficult to put these bearings on, whether it is just due to ignorance or not, I don't know, but we have only ordinary mechanics, not any better than you have, in the steel industry. We do think that we can concentrate our mechanical ability at one place.

Not very long ago I had occasion to talk to an assistant electrical superintendent who has been encumbered with the maintenance of motors and the personnel. He said if his company would permit the installation of ball and roller bearings, he didn't know which was best, that he would guarantee to reduce the maintenance force in the electrical department 60 per cent. Well, he didn't have anything to back it with, merely a statement. He was a man with years of experience and I believe he knew what he was talking about. Possibly his 60 per cent is a little bit high. You can make your own deductions. Certainly, when you consider that we have cranes and machinery which is inaccessible, due to its operation for 24 hours a day, or any other machinery that must be stopped in order to lubricate it, and think of what these gentlemen have said about operating them for six months or a year without looking at them, that would cut off practically all of the maintenance on the modern mill motor, or the ordinary stationary motor. They do not require much attention otherwise. We find it practically all bearings. If we can maintain the bearings we don't have to worry about the windings or anything else. If you have studied the brushes you will find that they will outrun the bearings. It becomes a question, therefore, in our experience, of the bearings.

In installing the bearings it seems to us that the motor manufacturer or the bearing manufacturer, possibly both, have got to give different data if they care to have the users install bearings. We have tried to go by the data which the bearing manufacturer gives, which so far we believe is very good. On the other hand, it doesn't seem that we have the necessary information to do our proper calculating from the motor manufacturer. That will have to be produced. Eventually, it will be similar to other data for figuring brake torques and such as that.

‡Superintendent, Electrical Department, Corrigan & McKinney Company.

I'd like to ask the gentlemen who presented the papers where they advocate, if there is a definite point, a change from roller to ball bearings, or vice versa. It is our experience as the gentleman from Westinghouse brought out that where we get in the slow speed, heavy loads, that rollers are better than balls, although we have some comparatively slow speed and rather heavy load bearings working in the ball type. It is rather surprising what some of those ball bearings will stand in the way of abuse. It surprised us.

It is our belief that speed, that is r.p.m. is a very unsatisfactory term, and that a better way of expressing this factor is feet per minute.

T. V. Buckwalter*: I took note of a few of these questions as they came up and shall endeavor to answer them:

The first one I have is in connection with the press fit allowance for bearings on shafts. I want to state right at the start that the fitting on taper roller bearings and on ball bearings is rather divergent. We like to have the cups rather tight. On a rotating element, such as this shaft, we like to have the cones tight also. If this shaft was stationary and the housing was revolving such as we have on the front wheels of the automobile we would want the cone loose so that it would creep and distribute the wear. On an element like this we like to see the cone a tight fit and we don't ask the manufacturer to work to exceedingly close limits and we don't work real close ourselves. We think that a thousandth of an inch is close enough for commercial manufacturing in large quantities. On a job of this kind we would ask for a press fit from, say, one-half to two and one-half thousandths or possibly one to three thousandths. If that cone has a press fit of three thousandths it probably compresses the shaft a slight amount and expands the cone a slight amount, but that does not make any material difference. If the cone is increased a thousandth of an inch in diameter that means probably five thousandths more or less in making the allowance for adjustment, and that is all done automatically in making the assembly, and we think it is the economical way of assuring that you get a press fit. By this press fit of the cone on the shaft you eliminate any necessity for clamping devices to hold the cone or inner race on the shaft at the pinion end or the other end, and also no necessity for clamping devices on the cup.

We generally do not consider it necessary to have the cup float on a stationary cup mounting. Probably the largest usage on a stationary cup is on differentials of automobiles and there we purposely make them a tight fit. They have a sufficient amount of surface and the surface is working at a favorable advantage, being on the inside bore, so to speak, that the wear on the cup is practically negligible.

The question of speeds brought up several times, particularly the speed at which change is recommended from roller bearings to ball bearings. I don't know whether I can answer that. As far as I know there is no limit to the speed at which we can run either type of bearing, but I will say this: At high speeds and by high speeds I mean 3600 and over, we must consider that any lack of true in the bearing, such as eccentricities of the cone or of the rollers or any of the rotating parts on bearings carrying a large armature, which in this case illustrated weighs possibly 500 pounds, would mean that lack of true would cause the bearing to oscillate or wiggle that large 500 pound armature and that starts up severe vibration in the electric motor that would be negligible in the automobile where comparable power is transferred through a small bevel pinion weighing possibly four or five pounds and where a lack of true of two thousandths is not a vital matter.

The matter of speed is largely a question of how accurate you wish to make the bearings and pay for them. I don't think any limit can be set. I have not discovered any. Some of the manufacturers in Detroit where they push production on grinding to the 'nth degree have been using taper bearings as well as ball bearings well in excess of 10,000 r.p.m. One of the largest companies admit that the bearings don't last very long but they last materially longer than any plain bearing they can get to stand up, and as a result effects an economy in production.

In the matter of friction of oil that has been well covered. Mr. Burns has stated pretty well that the friction of the bearing depends to a very great extent on the viscosity of the oil and heat, and consequently, the loss in rotating the bearing is largely a function of the viscosity of the oil. I can't say that I am willing to state just where the line should be drawn between grease and oil lubricants. Certainly, the greases have been wonderfully improved in the last few years. We have found it advisable, in fact necessary, to work up an approved list of greases. In starting we engaged one of the best Standard Oil chemists that we could find. We not only saved many times his salary in the oil question but it has given us some good specifications for grease for general anti-friction bearing use.

I am sorry to relate that at the beginning of this work we couldn't get some of the big oil companies to furnish a grease or recommend a grease that would meet our ideas of a good anti-friction bearing lubricant. The smaller companies came across very favorably. I am glad to say now that the larger companies are seeing the light and are coming across with some better lubricants. My own belief is that there are probably at least a hundred to one variation in quality in standard lubricants on the market.

I don't know whether you would be interested in this fact but some of our experience indicated that with certain lubricants that have been developed there is no metallic contact between the roller and the races. We have proved that by practically insulating the armature from the field with a few lubricants that are available, where as the ordinary lubricants that you can get in the gas stations would probably pass the current through a bearing just like a short circuit. This demonstrates that some lubricants have that property of maintaining the oil film even under the abnormally heavy unit loads that Mr. Gurney has brought out as occurring in anti-friction bearings.

I would certainly be interested in getting that fellow that wields the sledge that hammers bearings so that they run cool after they have had a spell of heating. I can think of many instances where that man and his skill would be very valuable.

The question has been brought up as to the matter of publishing data on bearing capacities. I might state that our company sold probably 100,000,000

*Chief Engineer, Timken Roller Bearing Company, Canton, Ohio.

bearings and we have never published any data on bearing capacities. Most of the bearings are used in tremendous quantities. One company uses upwards of 30,000 a day, and if they work out the proper selections once they are not interested further in what the carrying capacity is on paper, as long as it does their work. We are rather of the opinion that what the user of the electric motor is interested in is the bearing that stands up, and he doesn't care whether it is rated at so many pounds, so many revolutions, or what rating it is. Our experience changes. We might publish a rating today and find that next year it ought to be changed somewhat, and if the ratings are already published there might be some hesitancy in changing them. On the other hand, the engineer working on the mechanical problems likes to work out his problem to a finality, to a greater extent than the automotive engineer. The automotive engineer leans on the bearing engineer for his bearing selections. He sends out his problems, but he doesn't make any effort to select his bearings. He thinks that is the duty of the bearing man, because the bearing man must in the last analysis take the responsibility for their successful operation. But if the mechanical engineer on the mechanical problem is going to insist on working out all of his problems including bearing selections I guess we will have to fall in line and give them the dope we have.

One of the steel experts brought out the matter of the facility with which bearings are serviced, and I believe this has been carried further on both ball and taper bearings than it has on plain bearings. I know that our company has probably some 32 branches, and probably 2500 stations where bearings can be serviced. In addition to these there are the service stations of the automotive and other customers.

One of the questions brought up was on the split bearing. Mr. Gurney very well stated the necessity of the steel being free of seams and perfectly homogeneous. That practically precludes the possibility of making either the races, the cup or the cone, split. I think we will have to adhere to the practice of removing the cup and cone as a unit.

Another speaker brought out the matter of the introduction of bearings in wood working machinery, working at high speeds. I am glad to state that this is one field that has been very well handled by the taper roller bearing, and also I think the ball bearing. Some of the letters we have from wood working machinery manufacturers have been very flattering.

D. M. Petty*: This is the first opportunity I have had to visit the Chicago Section. I am certainly glad I was able to get out at this time.

First, the general scheme of ball bearings, something about how this all came about. Mr. Hess, of Youngstown, started the ball rolling several years ago by applying roller bearings to mill motors, and reported good results. A great many of the rest of us had used ball bearings spasmodically and intermittently, where we knew we could not get away with other kinds of bearings.

The results of these various kinds of bearing applications were so satisfactory in the majority of cases that a number of us felt that the motor industries, as far as the steel mills were concerned, would eventually go to ball and roller bearings.

So as to avoid the complications which we have found in other developments of electric motors, for instance, the mounting dimensions of the mill type motor, so that no two manufacturers would have a motor of the same hp. and speed, which would have the same mounting dimensions—so as to avoid that same "Comedy of Errors", if you want to call it that, we thought we'd step into this anti-friction bearing game early enough so that we could at least have some degree of standardization, so that the bearings as laid out by one company would not be entirely dissimilar to those laid out by another.

The Bearing Committee, as it has been called this year, of the Association of Iron and Steel Electrical Engineers was originally the Motor Committee, and we have been discussing and considering this subject for about three years. We could see from our own conversations among ourselves, and from conversations with the bearing manufacturers, we could see no reason why a line of bearings could not be standardized upon, and why we could not say to the Power Club, or to the Motor Builders, that this is the line of standard bearings for motors and none others shall be used.

Our first conversations with the Power Club, which, as you all know, represents the motor builders, we were received very coldly. However, we kept plugging and swinging the Big Stick, until this year our conferences have been very satisfactory.

At the Buffalo Convention the bearing engineers were good enough to come up in force, and we got them around the table with the motor designers, and most of the troubles which the motor designers had suggested to us, that this could not be done, and that could not be done, melted away under the bright sunshine of the bearing engineers.

We laid out a set of bearings at Buffalo which, as Mr. Reynolds spoke of as the 300 series, we told the motor people we saw no reason why the 300 series could not be standardized upon as the line of bearings which should go into motors.

I believe that opinion was rather unanimous at Buffalo, but when it went down to the Power Club certain objections were found. So that on Tuesday and Wednesday of this week, January 8 and 9, 1924, a joint meeting of the Power Club's Committee and our Committee, Mr. Hess and myself representing the Association, met together again to find out what these troubles were.

After discussing them for two days we found that they were mostly imaginary, so that apparently, in a very short time, a definite line-up of standard bearings for motors will be announced.

Since this is a problem involving the motor builder primarily, we feel that the Power Club should make the announcement as to what it is. All I can say at this time, or all that I care to say, is that it will be the 300 series with additions, which are still within the 300 series, however, that is, involving what is known as the 300 series extra wide bearing or heavy, I guess, extra wide bearing is the proper term for it.

This has the advantage that the roller bearings, as Mr. Batesole pointed out, in the 300 series, have essentially the same mounting dimensions, that is, the inside bore and the outside bore, for the same number

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of bearing, is the same. So that, if you have a motor in your plant that you buy with ball bearings, and you find later on that the bearing on the business end of the motor is a little light, you can slip a roller bearing into its place, and get additional capacity.

In general, it appears that ball bearings are proper for motors whose speeds are high and whose load is not too high, and that roller bearings are proper for motors whose top speed is, say, 1,000 r.p.m., and whose load is rather high. In general, that is apparently the situation as it stands today.

I am glad to say that practically all motor builders who have attended these conferences have now changed from the standpoint of being rather cold to the subject, and are now rather enthusiastic on the subject, so that I don't believe, in the near future, you will have any trouble buying all the ball or roller bearing motors you want.

My own experience on anti-friction bearings has been limited to the ball bearing. I purposely avoided going into roller bearings until after this standardization question was fairly well settled. I had to go into ball bearings in a number of cases because I could not operate certain motors that I had any other way. The sleeve bearings, no soft metal sleeve bearings could be put in that would stand the racket.

At this point, from my own fears in the past, and from what I have heard said by other steel mill engineers, they have been inclined to doubt the capacity of ball bearings to carry shock loads. For instance, if one tooth on a pinion was a little over-sized that hammer would soon knock a ball bearing all to pieces.

It just happens that the places where I have had to install ball bearings were places in which I could not keep a sleeve bearing, just because of that shock load. We had one or two cases of that kind, where the average life of a sleeve bearing was about three weeks. We had one particular pet case of that kind where we have a ball bearing in service in identically the same motor, under identically the same conditions, which is now going into its third year of life, so that we believe that there is no danger in using ball bearings under shock load conditions.

I think the same thing will remain true under shock conditions and conditions which have no shocks, such as direct connections, that probably the ball bearing will last longer with the direct connections than it will with the shock loads, but the relative advantages of using ball bearings with shock loads is just as great as it is any other place.

I was particularly interested in what Mr. Batesole and what Mr. Reynolds had to say in reference to lubrication. When we started studying this problem of anti-friction bearings, one of the things we did was to find out from a large number of steel mills who kept accurate records, as to the causes of their armature failures. Investigation into those causes showed us that a very high percentage, somewhere in the neighborhood of 80 per cent of all armature failures, was caused by oil which should have stayed in the bearings, which did not stay in, so that one of the principal arguments, and I think, the principal argument for the use of anti-friction bearings, as a whole, is the elimination of oil.

I think it would be rather difficult for any of us to show a dollars and cents saving in bearing cost over the sleeve bearing, as compared to the ball bearing. For instance, if we will take any particular application, if the life of the sleeve bearing, average life of the sleeve bearing, we will say, has been a year, the average life of the ball bearing would have to be many times that in order to show a saving from a cost standpoint of renewal of bearings, because there is no question but what you can take a bearing shell and rebabbitt it, bore it out, at a cost much lower than the cost of putting in a new ball bearing. And generally speaking, ball bearings cannot be repaired. They are put in new and they are thrown away when they are worn out. That is about all there is to it. I understand there is a possibility of taking a bearing which has been worn, as Mr. Reynolds stated, due possibly to outside matter getting into the bearing, and a lapping occurring in the bearing, so that it has become sloppy, it is possible, I understand, to take those instances, take those inner and outer races and regrind them and put in over-sized balls. I don't believe I have had any one state to me definitely whether they thought that was a paying proposition or not. So that, generally speaking, we can say that ball and roller bearings can't be repaired. When they wear out you throw them away and put in new ones, so you have got to get an exceedingly long life, and I believe, personally, a longer life than you can expect from the ball and roller bearing to justify on a cost basis, thinking of the bearing only, but you don't have to rewind, you don't have to save many rewindings of armatures, in order to pay for a set of ball bearings, so that if you can eliminate half, and I believe you will eliminate more than that, but if you do eliminate half of the number of rewindings that you would normally have, using sleeve bearings, by using ball bearings or roller bearings, I believe you have paid a very handsome dividend on the cost of those bearings.

The Bearing Committee of the Association has arranged a series of meetings, this being the second one. The first one was held last Saturday night in Philadelphia, and there will be one next Monday night in Clveland. The fourth one will be in Pittsburgh on the nineteenth, or a week from tomorrow night.

We have had a ball bearing man and a roller bearing man at each of these meetings. We hope that all of the operating engineers who have had experiences, either in the question of lubrication or mounting or any of the other pertinent phases of the subject will give their discussions, and we hope that we will get and collect all of these papers together, along with their proper illustrations and publish them in one of the monthly issues of our regular journal in due time. This will probably be held open until the time when the Power Club is ready to announce their acceptance of the work done at Cleveland this week, so that when that comes out, I think all of you will have a booklet on the subject of bearings that will have a considerable degree of authority.

One of the remarkable phases of the work which has come to my attention on the ball bearing and roller bearing question, from the engineers, starting at Buffalo, has been the remarkable way you might consider two competing organizations have worked together, the ball bearing engineers and the roller bearing engineers. I think they both feel down in their hearts that their own particular apparatus is a little better than anybody else's, but to say the least, they are very polite as to how they say it. It has been a pleasure

to me to hear all these engineers, representing the different companies, speak of their own bearing, their own product, and in the same breath compliment some one of their competitors. It makes me feel they are a set of men in whom we can have a great deal of confidence. I believe they all think the other fellow has a pretty good product, and possibly way back in their minds possibly they believe the other fellow has a little something on them in one or two points, but we can't blame them if they don't admit it.

L. Hess*: It gives me a great deal of pleasure to be able to meet you and talk to you this evening. I wish that Mr. Hall had asked me to talk a little later, as I would have preferred to have heard some comments from you men as to what you thought about the whole subject first. It is a pet of mine, of course, and there may be a good many points brought out here later on which I could throw some light.

You are undoubtedly familiar to a large extent with my experience, and I won't go into that, as it was pretty well outlined in my paper which was given in Philadelphia two years ago, but there are a few points in connection with the papers that have been given tonight that I wish to bring out.

On one or two of them I disagree slightly, from a practical stand point, with the engineers who have talked to you. The matter of end thrust has been brought up and designs have been shown, and in one of the papers, designs were recommended, whereby a ball bearing is used in one end to act as a thrust member, and also to prevent end motion.

Now if a motor runs constantly in the same place, even though it is in the correct magnetic center, the tendency is to score the commutator, due, I should say, to no side movement under the brushes. For that reason on our larger generators we put oscillators; in our smaller motors we allow some end play in boring our bearings, so that the natural tendency of the motor under any sort of a drive is to oscillate slightly and find its magnetic center. It is not always possible to machine the parts of a motor so that the magnetic center will be exactly the mechanical center of the motor, and if the armature is out of magnetic center longitudinally, there is a constant tendency for it to thrust back against the bearing which is opposing its reaching that center.

We have found in our practice that it is better to allow a floating action to take place, and I believe that the best arrangement to do that in the case of the roller bearing, and I am talking largely on those, because that is where my experience has been, is to make the outer race of the bearing slightly wider than the rollers, so that while the outer race is locked in the housing, and the inner race is locked on the shaft, there is a chance for the armature to float on the rollers between the two races. We put the wider race on the outside because of its larger area, since there is a certain wear which will take place from the sideways slip of the rollers, and we would like to have as large a surface for that wear as possible.

We have found in our experience that with the ordinary type of roller bearing, and in the ordinary type of drive, such as a belt or a pinion, where there is no extraordinary thrust, the use of a thrust bearing is not necessary. Where a bevel drive or something of that sort has to be contended with, of course, a special thrust bearing is necessary. But for ordinary mill drives, such as tables, screw-downs, and similar equipment, pinion or coupling driven, the use of end thrust bearings is superfluous. We used them in our first bearings because we were afraid of them. Examination of those bearings showed that the thrust bearing was doing no work, and in our later applications we have abandoned them.

The type of bearing which we use has two races, both of which are a sort of D or channel section, and the rollers riding between; the space where the rollers are is completely occupied and there is no retaining cage.

As Mr. Petty said, the first thing that showed us the necessity of the anti-friction bearing was the oil problem, and it was with the view of eliminating the oil that we went to the anti-friction bearing, where we could use grease. We have had a good many discussions on the use of heavy oil and grease, but at Youngstown we have stuck to grease entirely, and we have found that it works out satisfactorily. We use a semi-fluid grease of about the specifications that the papers have made tonight, and we have found the same situation as one of the speakers mentioned, the fact that two barrels of grease from the same company, under the same brand, are not identical, and in order to overcome that we have set aside certain barrels of grease to be used in motor bearings, and a sample from each of those barrels is sent through our laboratory for analysis to be sure that we have the same product. The presence of any fibrous material in a grease, any filler, talc, graphite or anything of that sort is ruinous, and it will pack between the rollers and eventually freeze the bearing, as the speaker said.

The question of greasing is, to my mind, the most important thing there is in the operation of roller bearings. I am not so fearful of dirt. We have run them under all sorts of dirty conditions, and the grease in the bearing gets black in a very little while. There is no perceptible wear in two or three weeks or a month in a roller bearing which would cause small particles of steel to blacken a grease, yet in steel mill work we find, inside of a month, grease put in a light yellow color comes out black. The only answer is that dirt has gotten in there. We don't find that the dirt is particularly harmful in the quantity it gets into the bearing.

I have, running in my plant, two sets of bearings which about the first of May of this year will be going into their seventh year of active service. Those two particular bearings have been very carefully watched. They are on size EW Crocker Wheeler Motors on bridge service on a mixer crane. They have not been in absolutely continuous service. We have had some electrical troubles on the armature which have necessitated their being changed and repaired, but we have kept them out of service the minimum amount of time necessary to make the repairs. To date the bearings are operating as successfully as they ever did. I will not attempt to predict the life of them. This is a slow speed drive, and is not to be compared with a thousand revolutions per minute, but at the same time it is a very heavy shock drive, and under the worst dirt conditions I believe we have in the steel industry.

I don't know that I can add anything further. If Mr. Hall opens the subject for general discussion,

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later on, and anybody has any questions which I can answer, I will be very glad to do so. I feel very strongly that this whole subject should be as widely discussed as possible. Everybody's experience should be found out and published, so that we may all know. My experience has been confined almost entirely to one type of roller bearing, and to heavy duty work, and comparatively slow speeds, never in excess of a thousand revolutions per minute, on cranes, tables and general mill machinery, and we have had practically no failures. In fact except for a case of a bearing which I ran to destruction on purpose, we can say we have had no failures in bearings, and I have in the neighborhood of 300 sets in operation today.

J. M. Parsons*: There is very little that I have to add to what has already been said. The subject of ball bearings and its application to motors has been very ably covered by the previous speakers.

There was one phase, however, that I would like to speak of, that has to do with the handling of a bearing when it is being assembled in the motor.

A bearing that comes from the ball bearing manufacturer has been very thoroughly cleaned. It has also been very thoroughly slushed in a rust-preventing compound, and that compound is also a very excellent lubricant for the bearing. I think that is true with roller bearings as well. When the bearing is to be used it should be drawn out of the stock and taken right direct to the work in the package, and should not be removed until ready to use, and it is not necessary to clean a bearing.

If, in handling the bearing, it has become dirty and it is necessary to clean it, the best cleaning medium is kerosene. Gasoline has a bad effect because it dries rapidly and generally has a chemical action in connection with the grease that is on the bearing, leaving a residue in the bearing that is destructive to it. So that I would caution users of bearings to exercise that degree of carefulness in handling a bearing, to see that the bearing is not exposed unnecessarily to dirt and to bring it to the works in the package in which it is packed.

It is not necessary to inspect the bearing. The bearing comes from the factory correct in size, and is ready for installation in the particular job that it was intended for.

I would like to ask Mr. Reynolds if his experience has indicated which bearing, either the in-board or the out-board bearing, should be confined between the lips of the housings?

H. Reynolds: Our experience in the motor application is confined almost exclusively to a.c. motors. We don't have the commutator trouble that Mr. Hess mentions. Under those conditions we invariably recommend the confining of the bearing opposite the pulley.

The bearing sizes for electric motors, as we all know if we have ever analyzed the forces, no matter how we see it, we will agree that the out-board bearing has very small load, so small a load that a 200 series bearing would calculate as satisfactory. At the same time, stock considerations and mounting considerations almost invariably lead the user of the bearings to use the same size of bearings at both ends, which would lead the bearing to take the thrust. Our practice and experience indicate that is right.

*U. S. Ball Bearing Company, Chicago, Ill.

R. W. Drake†: I stand in the same place that Mr. Petty has described. I have used ball bearings in some drives where I could not make anything else live. We have one drive, all of the same type, 36 motors, running in falling sand. We obtained an average life of six weeks with sleeve bearings. Reliability was an important point. Ten men were out of work whenever there was a failure on any one of three motors. That is, there were three motors on each machine. A breakdown on any one of the three put the gang out of work.

Under such circumstances the difference in cost of babbitt-lined sleeves and anti-friction bearings becomes trivial. We have found, however, that the cost of a new babbitt-lined sleeve, plus the cost of putting it in, overtime is greater than the cost of a ball bearing.

Several problems presented themselves to us in making our design. We were obliged to build the greater part of our own motors to get a design which would live, or rebuild them, rather.

One of our problems had to do with end thrust. There is no appreciable end thrust in operation, but we worried about what would happen when a man drove on a pinion. Under conditions where 10 men are out of mork when there is a motor breakdown we make an effort to give five or ten minute service. You can't ask a man to press on a pinion in five or ten minutes. We knew they'd be driven on with a 12 pound sledge. We wanted a design that would stand that. I am very hopeful that Mr. Hess may have something to offer on that subject.

We have solved the difficulty for driving on a pinion or coupling but theoretically, looking at our blue prints, it appears that we have not solved the difficulty of driving it off again. We ordinarily pull pinions with a screw. Our men are ordered to do that, but we know that in two cases pinions have been pulled by driving two cold chisels between the housing and the pinions, and the motors are still running with the same bearings. We are using bearings of the deep groove radial type.

We attribute our success to the fact that we based our first design on five years' experimental work done by Ernest Wanamaker, of the Rock Island Railroad, in another very severe application, axle driven generators for lighting pullman cars. When he began his experiments these generators were fitted with bronze sleeve bearings, and he said that only one trip to the coast and back was expected of a pair of bearings; that they never attempted, no matter what the condition of the bearing when it returned, to ask a pair of bearings to make a second trip to the coast.

It took him five years to work out a satisfactory design. His bearings also were lubricated with grit, we can realize the amount of grit pulled up by the eddies under a pullman car, passing over cinder ballast.

We adopted his method of dust seal, with further modifications. We elaborated it. Our motors often have sand piled on top of them. We have had absolutely no trouble and no apparent entrance of dust into the motors. We get some blackening of the grease, as Mr. Hess has noted, but we have always felt that that came from the wear of the felt packing rings, which we use, in addition to grooves.

†Electrical Engineer, International Harvester Company, Chicago, Ill.

Mr. Wanamaker, in speaking to us when we started our first design, told us, "Use bearings large enough. On this point, don't pay any attention to what the bearing salesmen tell you. Take what they say and then multiply it by about four." We are using a bearing on a three h.p., 1200 r.p.m., a.c. motor under extreme shock conditions which is rated by the manufacturers at approximately 50 h.p. for this speed. They drive what is substantially a fan with one paddle, which fan is paddling sand. This paddle wears out in six hours on some applications and 12 hours on others. Toward the end of its life there is a tremendous unbalance. The paddle is counter-balanced at the beginning. When worn out we have a pounding almost equal to a pinion with one tooth gone. Perhaps it is due to the enormous oversize and excessive capacity on that bearing that we are able to wedge off a pinion. We have no other explanation to offer.

We have been puzzled to devise an assembly without the use of thrust bearings which we wish to avoid, if possible, which will not offer the difficulty of expansion, and will give us a thrust in both directions.

It is difficult to explain the method we are using for taking thrust, without a drawing. In the front end bearing housing we provide a surface to limit the axial travel of the outer ball race in the usual manner. We provide another surface for the inner race to thrust against. The location of these two surfaces is such that with outer race bearing against its limiting surface, there is normally 0.005 in. axial clearance between the inner race and its limiting surface. The bearing manufacturers tell us that a bearing of this size will stand five-thousandths axial deflection. When we drive a coupling onto one of these motors the inner race springs over and strikes a solid surface. In that way we get something to take the blow without injury to the bearing.

The other difficulty which we anticipated was the difficulty of the ordinary type of motor inspector dismantling and disassembling a ball bearing assembly, especially in a place where sand was falling.

In our later design we have taken care of that by building our motors so that the ball bearings and their enclosing housing, are a unit with the rotor. That is, you can take off the end bells of a motor without disturbing or disassembling any part of the enclosing housing or dust protection, and you can handle rotors, complete with both bearing assemblies. Some of our motor inspectors have never seen the inside of the bearing enclosure. They take the rotors out, put the heads back on, and take the rotor with the two bearing assemblies into the shop.

These 39 motors have been in service from one to three years.

To date, with the exception of the one trial bearing of an inferior type, we have never had a bearing failure. We have never had to open a bearing assembly because the bearing failed. We have had to open the bearing assemblies because the shafts broke off or the key way was chewed up so that a new keyway was necessary.

James Thomson*: This discussion has been very interesting to me. I remember about six years ago, shortly after I came to this district, having a lot of trouble in our plant with plain motor bearings. There has been something said tonight about motors in cement plants, but I believe that the motors in the steel foundry are in just as bad shape, up against it, just as badly as the motors in a cement plant, because there is constantly being floated through the air a very fine sand which settles on the motors, and eventually gets into the bearings.

We happen to have nearly all our cranes equipped with Westinghouse Type K motors, and I can hardly imagine bearings worse for a steel foundry than those furnished by the Westinghouse people on Type K motors. We got, generally speaking, a life on those bronze sleeve bearings of from two to four months. In a steel foundry the conditions are pretty nearly as bad as in a steel mill in regard to keeping the cranes operating every minute of the day. In some particular departments it is even worse. So, about six years ago I took the matter up with the Westinghouse Company, to see if I could not get a little relief, on this bearing question and I don't think I'd care to tell you the answer I got, but in short, it was that I did not know what I was talking about when I mentioned ball bearings for a crane motor—absolutely absurd, why, the shock of reversal would break up the balls, and a lot more bunk!

We went along for a couple of years more struggling with this bronze bearing problem and I was getting more and more disgusted. Finally, we just took the bull by the horns, as it were, and made up one set of ball bearings for one motor.

This motor was placed on a bridge drive, in one of the worst places in the plant. After 20 months we took it down to examine it and found that it was still all right. That is just five times the life we could get from the best sleeve bearings in that same place, so we decided to go furhter with the experiment.

Thinking perhaps, that I did not know very much about ball bearings, I took the matter up with our friend, the Fafnir Company over here. They gave us their idea of design, and that coupled with our own ideas and experience, has enabled us to develop for some of our "K" motors, ball bearings, which are giving us good service. However, we did as Mr. Drake has suggested. We took Fafnir's recommendation and multiplied it two or three times, and we think we have a real bearing. If the Fafnir representative wants to know what it is like I will tell him after the meeting is adjourned.

That original experimental ball bearing motor is still running on a crane. That is three years and three months service right now, and in a place where we could not get more than about three or four months' life out of a sleeve bearing.

I was glad to hear some one else say that the cost of a bearing hardly enters into the matter at all. If a motor is down for a rewinding caused by low bearings, that, in addition to the cost of the sleeve bearings which you would put in at that time, practically pays for a set of ball bearings.

Well, that is only one phase of the question. The other is the fact of keeping your machine in operation, or at least, ready to operate all the time.

I have little more to add at this time, except that I am an enthusiast on the subject, and have been experimenting quietly for quite some time. I did not expect to do anything here tonight except sit and listen and learn, however I am pleased to say that we

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are having good success with ball bearing motors so far.

S. Farrell: I feel that the ball and roller bearing subject has been very ably covered. I might add in connection with Mr. Thomas' remarks, that the Rollway Company has developed an interchangeable housing for the Westinghouse Type K motors, and have several hundred in operation at the present time. I feel that in talking about these bearings that quite a lot of stress has been placed on the care and cleanliness of ball and roller bearings as applied to motors.

Now, we have found in applying bearings to motors—for instance in cintering plants, you gentlemen are probably familiar with cintering plant practice:

This is a product that is very abrasive, about 60 per cent iron. We furnished bearings for a company three years ago for use in their cintering plant. At that time they requested a felt seal on each end of bearing housing. The felt seal did not wear very long.

It was my privilege recently to examine that bearing, and I feel that with the lubricant which was in that housing I could easily have ground the valves of my automobile, and yet the condition of bearing indicated that they are good for much longer service.

In talking about fits for bearings, I might cite an example of a bearing, which has been applied to a motor driving a reversing table on a 40-in. blooming mill. That probably is a service with which most of you gentlemen are familiar. These bearings have been in service for over a year. I would say that the motor has probably 18 or 20 reversals in breaking down the ingot, probably taking a period of a minute or a minute and a half. These bearings are mounted without locking nuts. They are driven up against a shoulder on a shaft. It was the ordinary mill type motor and they turned the shaft down to a diameter suitable for a standard bore bearing. This shaft was machined about one and a half ten-thousandths larger than the bore of the bearing, and bearing was driven on with a block of wood and a hammer.

Recently, after it had completed its year's service the engineer in charge talked about opening it up to see what it was like. But later decided that as long as it is working all right, they would take it out for inspection.

Those are two points that I think that most of you gentlemen are interested in: One, to have a bearing which can be applied to all services, and secondly a bearing which can be installed in present motors without assistance of expert machinists.

Now, when you think about the reversals which this table motor had, there would be a tendancy for its bearing to move on the shaft, as I noticed before the motor was changed from a babbitt bearing to roller bearing, that there were about five-sixteenths or three-eighths of an inch oscillation of the armature probably caused by currents in the motor or the misalignment of the pininons or the gears which meshed into the motor pinion. The bearings used have a channeled section, top and bottom, races and the armature is tied permanently in its field. Oscillation thrust is taken on channel sides. We only permit about five or six thousandths for shaft expansion.

As far as commutation is concerned, that has never worried this engineer. He appreciates an inner pole motor and uses a good grade of brush and to date armature is working without ridges on its commutator.

One of the particular reasons for changing this armature from a sleeve bearing to a roller bearing was the common cause of oil splash. Every time that armature reversed, which I said was about 18 times in a period of a minute or so, there was a vapor of oil came from the shoulder of the shaft which lodged on the front of the commutator. Inside of two or three weeks the oil caused a bad mica in commutator or grounding of the front cone ring. The armature has been in continuous service for a year.

R. H. DeMott*: Mr. Chairman, the time is getting short. I do not know that I can add a great deal. We feel that the successful operation of ball bearings depends upon three essentials. First: Proper selection of the bearing for type and size. Second: Proper application of the bearing. Third: Proper care of the bearing.

I would like to impress upon the steel mill men who are interested in the application of ball bearings to electric motors that they have two problems. One—a changeover of motors in their own plants; the other a purchase of motors from manufacturers in position to supply anti-friction bearing motors. From the changeover standpoint you are faced with the question of bearing selection and housing design and it is our advice that you call in the bearing engineer and let him assist in the proper selection of the bearing and the proper design of housing.

Mr. Petty has pointed out that there will be a standardization of the series of bearings. Your problem will then be to obtain the proper size of bearing in that series to meet your conditions. In this respect do not let your Purchasing Department enter into the decision too strongly for their tendency might be to compare the cost of, say, a No. 305 with a No. 307 and decide upon the least expensive of the two and this might mean that the bearing might not be sufficiently large.

Your next problem is the design of housing which need not be complicated, as the speakers have pointed out. There are certain proven methods of keeping the lubricant within the housing and preventing dust and dirt from entering the housing and as experience has been our best teacher in this particular we ask that you permit the anti-friction bearing engineer to help you solve this problem also.

One anti-friction bearing engineer's opinion may differ slightly from that of another. Do not let this disturb you too greatly for it is the exception rather than the rule for two engineers to absolutely agree on any one subject.

The question of the care of bearings is something that is entirely up to the men in the mills and this question is largely one of lubrication.

In mentioning grease as a lubricant Mr. Reynolds cited an instance where one bearing had run six years without attention, whereas another ran only six months with the same lubricant and in the latter case the volatile substance of the grease had disappeared and nothing but a hard cake or residue left.

Our recommendation is that you examine the bearings every two or three months. It is not a difficult problem to establish a regular schedule of lubrication attention. One gentleman, I believe it was Mr. Hess, mentioned in Cleveland the other day that he had established in his plant what he called "Lubrication Monday". This meant that every motor was inspected and lubricated on the first Monday of each month and through a proper and systematic attention he was able to determine the quantity of lubricant needed and the problem of lubrication and maintenance was consequently very much simplified.

I feel that the Fafnir and Norma representatives should be complimented for the work they have done in

*Engineer, S. K. F. Company, New York, N. Y.

the preparation of their papers and I appreciate the opportunity of being here with you.

C. L. Newby*: Several points have come up during the discussions this evening that I believe citation of our experience will clarify:

The installations made applying our bearings to electric motors are slightly different than those previously described. The chief difference is that we have not found it necessary to consider interchange of the complete bearing because we find that it is possible to permanently install our inner race, sometimes known as inner sleeve on the armature shaft. We do this simply by shrinking it on, as mentioned earlier.

I agree with the last speaker that it is good practice, to design the bearing housing so that not only the sleeve which in our practice becomes a permanent part of the shaft, but the rest of the bearing as well as the bearing housing can be worked with as a unit in the mill.

In case of armature failure it will then only be necessary to consider the whole armature, including the bearings, as a single unit, that can be taken to the electric shop and whatever work is necessary done on it.

There has been some talk that would tend to create the impression that it is almost a watchmaker's task to install an anti-friction bearing properly. That is not the case. It is necessary to use ordinarily accurate means of measurement to get your shaft right to start off with, and use ordinary skill that is customary in any tool-room, as far as our experience shows, to get the bearings in place. But we don't believe, in fact, we know that super-accuracy is not necessary. We feel confident that the average mechanic or a mechanic that can be developed in the average steel mill shop can properly install anti-friction bearings.

We are also firmly convinced that a separate bearing for taking thrust is advisable and also that it is desirable to provide end flotation.

D. E. Batesole: I wish to call attention to one point brought out by Mr. Petty in connection with his experience with ball and roller bearings. He remarked that he believed roller bearings to be best suited to low speeds, say, up to about 1000 rpm., and ball bearings at high speeds. In this connection, it may be interesting to know that Hoffmann roller bearings have been used for years on electric motors operating at 3600 rpm. and on other applications they have been run at speeds up to 7200 rpm.

Referring to the remarks by Mr. Hess about the desirability of a slight end float of the armature on account of commutator wear, I wish to say that such a float would be possible and is sometimes used, with two Hoffmann roller bearings of the standard or separable type. In this case some type of locating surface is necessary to take care of endwise location. This point of course applies more to direct current motors. On alternating current motors we recommend the use of the deep groove ball bearing for positive location in each direction.

H. W. Brown†: I feel as htough every point had been covered from beginning to end. I don't believe that much more can be said. The ball bearing salesmen have been humorously knocked; the engineers have been told they don't agree at all in their recommendations, etc.

One point was brought up tonight—that the compound that bearings are shipped in makes a good lubricant. It just happens that with our bearings we intend that the compound should be washed out with gasoline, and then the proper lubricant put in.

The proper lubricant depends upon the application of the bearings, the speed at which they operate, the closure, the temperature, of the surrounding air, and, its selection is a matter which should receive careful thought.

I am sure all the bearing manufacturers feel very much indebted to Mr. Petty in arranging for these four meetings of the A. I. and S. E. E. and for the meeting with the Power Club. Speaking for the New Departure Mfg. Company, we feel very grateful to Mr. Petty, of your Bearings Committee, in this matter.

H. C. Davis*: I think the hour is somewhat late to attempt to review the many good points that have been brought out. I will only emphasize one thing that Mr. Petty brought to your attention, namely this, the bearing manufacturers seek to co-operate with the users of bearings, to give a real service which is of money value to you. To that end we are constantly investigating new alloys in the steels; we are constantly breaking down bearings in our own laboratories, of our own and our competitors' makes. For that reason I think we acquire respect for our competitors' bearings.

We men who are in the ball bearing business are all of us young at that business. Probably we have had far more experience in other lines. For that reason we are able to appreciate the operating conditions which you men have to meet. We meet the same conditions ourselves in our own shops, using ball bearings under the same practical conditions, as have been described here, but I think if we were to describe our own experiences we would not be able to approach the recommendations for ball bearings which have been given by these engineers of the Iron and Steel Association.

W. B. Wachtler: I should like to ask Mr. Farrell whether the press fit which he spoke of, and which I understood him to mention as "a tenth and a half" should not have been "a thousandth and a half." This point is mentioned in order that the records may be correct in this important detail.

Another condition that has been brought out repeatedly this evening is the wide variation in quality of various makes of greases and of the same make of grease when different purchase lots are compared. It would seem advisable, in view of the good work done by the A. I. and S. E. E. in formulating standards of various kinds, including roller and ball bearings, that some committee be delegated to line up the lubricant manufacturers and their products in the same way.

From Mr. Reynolds' talk I assume that he favors locating the ball bearing which is restrained axially, at the light duty end of the motor. This is undoubtedly done to place the thrust load on the bearing which carries the lighter radial load. It has the disadvantage that it requires the bearing at the heavy duty end to slide in order to compensate for unequal expansion of frame and shaft. Instead of sliding, this bearing sometimes becomes slightly cocked in the bore. Consequently it has been found better in many cases to make the bearing at the heavy duty end large enough to carry both radial and thrust loads, and fix this bearing in place axially, leaving the lightly loaded and smaller bearing at the other end to take care of expansion effects.

S. Farrell: One and a half ten-thousandths—it was driven on with a block of wood and a hand hammer. The width of that bearing is the width of a double race ball bearing, but the outer width is the width of a single race ball bearing. Is that clear?

*Hyatt Roller Bearing Company, Newark, N. Y.

†New Departure Ball Bearing Company, Bristol, Conn.

L. Hess: The answer is, that we don't do it. We don't take them off out in the mill with a sledge hammer for that is absolutely prohibited. Just as has been said, the armature is a spare only when it has its roller bearings on it, and we change all pinions and roller bearings or do any work that has to be done on the armature in the shop. Our armatures go out into the field, on the spare pile, with the pinions and bearings on them. They go into the frames without disturbing the bearings at all. They are assembled in the shop by mechanics who understand the principles of the work. We don't sledge pinions off if we can prevent it, even in the shop. We use a press and try to keep jars of that sort off the armature.

I am rather glad the subject of the Type K came up. That was one of the problems we had to face, the identical problem spoken of here, the wearing out of the bronze sleeve in the bearing. We developed a bearing in conjuction with Mr. Farrell's Company, which has cured the Type K of its troubles.

There is one thing I did not point out, which came up in the meeting at Cleveland last Tuesday. It is a little at variance with some of the statements made in the paper. I happen to have been in on some tests of roller and ball bearings that were made in one of the factories in Cleveland under high speed conditions, that is, motors running up to 1600 rpm. Several of these bearings overheated, and the cure-all for that is to take out most of the lubricant. That sounds strange, but it is the lubricant that is producing the heat. Almost invariably, they found when a bearing overheated, that the shop assembly man had packed that bearing full of grease. If nine-tenths of the grease was taken out, the bearings cooled off immediately. So don't over-grease your bearings. It is not necessary and only causes extra heat and trouble in the bearing.

James Farrington*: We have had very few of our motors equipped with ball bearings and none with roller bearings. The ball bearing motors are on locomotives in coal mines which, as you know, is very severe service. We had an average length of life of 15 weeks on phosphorus bronze bearings and changed over about six years ago to ball bearings and get an average life on ball bearings of about 23 months. We then bring those races in, regrind them and get a ball of about 1/16 or 1/32 of an inch less in diameter and get 11 months' more service out of them, and have had armature failures drop to 65 per cent since we have had ball bearings. I have motor operated pumps and I really believe anti-friction bearings are going to be a means of eliminating a great deal of the failures. We are now taking steps to put roller bearings on our screw-downs, which you know is hard service. If it stands up there, I, for one, will be thoroughly convinced it is the proper thing.

R. S. Shoemaker†: It is unfortunate that we do not have with us tonight more of our active members who have had practical experience with the application of anti-friction bearings to motors.

At our plants we have not as yet made any extensive installations; we have been waiting to see how successful some of the larger installations are going to turn out.

The question of ball versus roller bearings is, I believe, still an open question and can only be decided by extensive trials of both types, and it will then most likely be found that each will have its special field.

A. Jones*: My personal experience with ball bearing motors is, when the ball bearing is fitted right it can't be beat. But the biggest step for the advancement of ball bearing motors would be an educational campaign by the ball bearing manufacturers amongst motor manufacturers using ball bearings to teach them to get a better fit. The salesman of the ball bearing motor manufacturer will tell you what a ball bearing fit is, but the factory seems to be unable to produce that fit. The trouble usually lies in the outer race not fitting the housing tight enough, and the bearing soon turns and wears out the housing. The other difficulty seems to be locking that bearing in place when using locknut. If any of the ball bearing manufacturers have a good way to lock said bearing in place they should give that idea to the motor manufacturers so they can use it.

J. M. Hipple†: One of the most interesting features of this whole situation to me is the awakened interest in bearings, whether they be ball, roller or sleeve bearings. It reminds me very much of a situation that some times develops in civic affairs where some issue arises which really arouses the people, and the newspapers speak of a new civic consciousness. I think ball or roller bearings have succeeded in arousing a bearing consciousness. The mill engineers and other operators have been for a long time regarding bearings more or less as a necessary evil and now they see a ray of hope in the concentration of effort for improvement.

So far as the ball bearings themselves are concerned, we have had quite an extensive experience with them. We began using them about 20 years ago, and so far as we can see, the bearing that we used then, the standard Hoffman ball bearing, is just as good a bearing and just the same bearing as can be had today. I do not think that progress made in ball bearings has been in the direction of a better bearing, but more in the nature of additional experience in the proper application of such bearings. We have applied ball bearings on our standard special motors where our experience has indicated that they will give good service and are economically advisable. We have attempted to be guided by our experience and judgment as to what constitutes good engineering.

The question of sleeve bearings has been injected into the discussion. The question is whether or not sleeve bearings can be produced that will not leak oil. This same aroused interest in the whole bearing subject has led to developments in the sleeve bearing. While I do not want to say too much on that subject tonight, as we are not discussing sleeve bearings, yet I can assure you that such sleeve bearings will be available to you, and I believe the steel mill engineers are acting wisely in investigating this subject before going ahead with wholesale applications of roller and ball bearings. I think so far as the future is concerned, they should continue to be guided by the results of their experience and should make careful studies of the economics of the situation, in view of the sleeve bearing developments of which I have spoken.

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Lubrication of Anti-Friction Bearings in Electric Motors.

G. A. Unger*: The two papers read before the Pittsburgh Section of the Iron and Steel Electrical Engineers have duly pointed to the importance of correct lubrication of anti-friction bearings and although some of the information contained in the following may be a duplication of the papers read, the importance of the subject will be an **excuse** for it.

Having had frequent occasion to discuss with Mr. Brunner, the author of one of the papers, questions pertaining to the lubrication of anti-friction bearings, the writer has been invited by Mr. Brunner to take part in the discussion of the papers read before the Association on the subject of anti-friction bearing lubrication.

In order to obtain the full benefit of all the advantages of high grade anti-friction bearings, it is necessary to take care of the lubrication in a satisfactory manner.

Anti-friction bearings properly mounted and housed require very little attention, but it is essential that they are lubricated with the correct grade of lubricant, with the right quantity of lubricant and at proper time intervals. Between the established lubrication periods the bearings should be left **severely** alone.

Since the application of anti-friction bearing lubricants has to take place only at infrequent intervals and in relatively small quantities, it should be possible to employ only high grade lubricants and to protect the bearings as well as the lubricants from contamination by dirt, grit or water.

The question as to whether oil or grease should be used for the lubrication of anti-friction bearings in electric motors has been generally answered in favor of grease, as long as the prevailing operating temperatures are below the melting point of the grease available for bearing lubrication. The writer has recommended and used a special high melting point grease capable of operating without melting at temperatures up to 220 to 240 deg. F.

The preference for grease as a lubricant is due (1) to the fact that the mechanical design of the bearing **seals** is less complicated for grease than for oil. This, of course, means that grease lubricated bearings will not cause oil soaked armatures and will require less frequent replenishment of the lubricant and provide a more effective seal against the intrusion of dirt and grit; (2) that for high speed bearing applications the temperature rise with the proper grades of grease is less than for oil bath lubrication. An oil drop feed lubrication to avoid churning, would be ideal in theory but difficult in practical use because of mechanical complications.

Grease Lubrication.

The proper grade of grease to be used for electric motor anti-friction bearings will depend to a certain extent upon the bearing and retainer type employed. Experiments conducted at the laboratory of the Joseph Dixon Crucible Company checked by results obtained in service, have shown that a soft cup grease is the most satisfactory lubricant for the majority of bearing types. In other bearing types a semi-fluid grease is the most desirable lubricant.

*Lubricating Engineer, Dixon Crucible Company.

The requirements for a satisfactory anti-friction bearing grease are high and the following conditions should be fulfilled:

Minimum temperature rise in operation, indicating a minimum of internal frictional losses by proper lubrication of races, rolling members and retainers.

Unchanged consistency for lowest outside and maximum motor temperatures.

Absence of corrosive and abrasive matter.

Lubricant should not harden or gum in service.

Long life, i. e., unimpaired maintenance of the above properties for long period of time.

Grease Quantity.

The lubricant quantity applied to each bearing housing is of paramount importance.

The bearing housing should be designed so as to form an ample lubricant chamber **on both sides** of each bearing.

The lubricant chamber, however, should under no circumstances be filled entirely, and charge of greases from one-fourth to one-third of the chamber capacity will give the most satisfactory results.

Provided that the housing design is correct in the first place and the correct lubricant is used, no lubricant leakage will occur and therefore no intermittent additions of grease should be made. They would only tend to fill up the housing and cause heating and consequent leakage.

The principal requirements of the housing design for grease lubrication are the following:

1—Facilities to introduce the proper quantity.

2—Facilities to remove the old lubricant and clean the housing without dismantling the housing assembly.

3—Simplicity of design to accomplish the above.

In slide No. 4 illustrating Mr. Brunner's paper, a design fulfilling these requirements has been shown for a 30-hp. general purpose d.c. motor.

In referring to it, we see that the lubricant can be inserted by means of a grease gun, through the upper fitting. The opening is protected from dirt by means of a self-closing spring actuated cap. This eliminates the effects of human neglect always present with caps which have to be screwed on or otherwise fastened.

The proper lubricant quantity should be determined and clearly marked on the bearing housing. The lubricant injected will spread to both sides of the bearing, owing to the arrangement of the fitting. To avoid overloading or mistakes, the writer recommends the employment of grease guns of a capacity suitable for **one** charge of lubricant. Compromises in gun capacities will be feasible so as to avoid an unnecessarily large number of gun sizes.

The writer wishes to emphasize here the importance of Mr. Bell's remarks regarding the danger of overcharging bearing housing with grease owing to the employment of the existing types of lubrication pressure fittings and guns.

The cleaning out of the old lubricant is facilitated by the arrangement of the drain. The drain opening is connected to both sides of the bearing and is closed by a self closing spring actuated cap.

To clean out the used grease, inject with the gun a definite quantity of hot light oil (heated to about

200 to 220 deg. F.) through the lubricant inlet into the housing and spin the armature. Drain by opening the drain cap and if necessary repeat the dissolving operation with hot oil. Avoid the use of kerosene or gasoline. These solvents will readily corrode the bearing surfaces.

When cleaning out the old lubricant and putting in fresh grease, the felt washers or other packing rings employed to seal the bearing housing, should always be inspected.

The cleaning operation being completed, fresh grease is then injected as described before.

The design shown can, of course, be readily modified to suit a different type of motor. The fittings instead of being built into the endframes can be attached to a pipe screwed onto the endframe.

Frequency of Grease Renewal.

This will depend to a certain extent upon the size of the lubricant chamber and the nature of the lubricant.

To avoid complications, it will be advisable to change the lubricant every six months. A weeky check upon the bearing temperature and opening of the drain cap to see if any oil has separated from the grease, is all the attention recommended.

In order to insure the proper lubrication of the various motors operating in a steel mill and also to keep track of the cost of lubrication, the writer has found the use of a simple form of lubrication chart of considerable value. These Dixon Lubrication Charts can also be combined with a maintenance schedule of the motors, which takes care of repairs at definite intervals.

Storing and Handling of Grease.

The grease employed for lubricating the bearings should always be purchased and stored in metal containers to which suitable grease pumps can be attached. This will eliminate the possibility of dirt entering into the lubricant, which is invariably the case when grease is stored in open barrels or containers with removable tops. The employment of a grease pump will also greatly eliminate waste of the lubricant and provide the facility of accurately measuring the quantities, thereby providing an effective check upon the waste of the lubricant.

Instruction Plates.

The writer recommends that grease lubricated anti-friction bearing motors be provided with instruction plates containing lubrication instructions in condensed form. The following is suggested:

1—Use only grease approved by bearing manufacturers.

2—Every six months wash out old grease with hot light oil.

3—For dissolving old grease use.....pints of hot oil.

4—Refill each bearing with.....oz. of grease.

5—Inspect bearings seals and if necessary renew packing rings.

6—Do not add grease between the six-month filling periods.

7—Keep dirt and grit away from lubricant and bearings.

Oil Lubrication.

As pointed out before, it is necessary to resort to oil lubrication in such cases where the operating temperatures of the motor exceeds the melting point of the available high temperature grease, and in the writer's experience this is the case above 240 deg. F.

Requirements for a satisfactory anti-friction bearing oil are high and the following conditions should be fulfilled:

1—They must fulfill the Government specifications as given in the Bureau of Mines Technical Paper No. 323, for Class "A" lubricants.

2—A Saybolt viscosity of from 180 to 220 seconds at 100 deg. F.

3—A pour test at the lowest outside temperature at which the motor may operate.

4—Absence of corrosive and abrasive matter.

5—Oil should not gum or become sticky in service.

6—Long life, i. e., unimpaired maintenance of the above properties for long period of time.

For motors operating at temperatures below the bearing temperature of the steel employed in the bearings (approximately 320 deg. F.), the design of the bearing housing should then provide an oil bath lubrication with ample lubricant chambers **on both sides** of each bearing. This will prevent churning of the lubricant and therefore prevent unnecessary temperature rise and also reduce the danger of leakage. The oil should reach approximately the center of the lowest ball or roller.

To prevent overfilling of the bearing housing, an oil overflow should be provided, which will permit the oil to drain to the outside of the motor when the proper level is reached. This overflow opening should be protected against the intrusion of dust, by a suitable shroud which may be cast onto the bearing housing.

The principal requirements of the housing design for oil lubrication are the following:

1—Effective seal to prevent leakage of oil and to prevent intrusion of dirt or grit.

2—Facilities to introduce the oil and to prevent overfilling.

3—Facilities to drain the used oil and to clean out the sediment.

4—Simplicity of design to accomplish the above.

The subject of bearing seals will be discussed below in the paragraph dealing both with grease and oil lubricated bearings.

The oil inlet should be provided with a self closing spring actuated cap. This eliminates contamination of the oil by dirt possible with caps or closures which have to be screwed on or otherwise fastened.

This oil inlet can be suitably combined with the oil overflow opening.

Drain passages connecting to both sides of the bearing should be provided and the writer recommends the combination of the drain passages with a settling chamber for dirt and grit.

The drain opening should be closed by a suitable plug or fitting and to prevent unnecessary leakage and waste of lubricant, the writer recommends the employment of a spring operated check valve which can be readily lifted from its seat by a suitable rod or lever.

Frequency of Oil Renewal.

This will depend partly upon the effectiveness of the oil seal and also to a certain extent upon the size of the lubricant chamber.

The writer recommends that oil be added weekly until it overflows, in order to take care of losses by leakage.

The oil should be changed every six months. The old oil should be drained and the housing flushed out with hot oil, heated to about 200 deg. F., so as to remove dirt and sediment. After this the housing should be refilled with fresh oil.

In order to insure proper attention to the various motors operating in steel mills and also to keep track of the cost of the lubrication, the writer recommends, as in the case of grease lubricated motors, the use of lubrication charts.

Instruction Plates.

The writer recommends that oil lubricated anti-friction bearing motors be provided with instruction plates containing lubricating instructions in condensed form. The following is suggested:

1—Use only oils approved by bearing manufacturers.

2—Inspect oil level weekly.

3—Add oil until it overflows.

4—Every six months drain old oil and flush out sediment with hot oil. Refill with fresh oil.

5—Inspect bearings seals and if necessary renew packing rings.

6—See that oil overflow does not become clogged.

7—Keep dirt and grit away from lubricant and bearing.

Lubrication of Anti-Friction Bearing Motors Operating at High Temperatures.

Where the operating temperatures of motors are above the safe operating temperature of the anti-friction bearing, the only satisfactory method to employ anti-friction bearings is by means of an oil circulating system with suitable radiators for cooling the oil. Such installation requires only low oil pressures, but the oil quantity circulated through the bearings must be sufficiently large to keep the bearing temperature below the maximum permissible temperature.

The bearing seals used with this type of mounting must be of a satisfactory type to prevent oil soaked armatures.

For the circulation of the oil, centrifugal or gear pumps will be equally satisfactory. The oil pump can be driven preferably by a silent chain from the armature shaft extension and the oil circulation will therefore be automatically governed by the motor operation. As oil tank and radiator should be of ample capacity so as to take care of at least three months' operation without refilling. The lubricant should be drained and changed every 12 months.

In installations having groups of motors lubricated with oil circulation, one central oil pump will be found satisfactory to circulate oil through the different bearing housings.

Bearing Seals.

The retention of the lubricant as well as protecting the bearings against dirt and grit is of importance not only for anti-friction bearing motors, but also for sleeve bearing motors. The subject of proper bearing seals is of considerable importance, but it can only be touched upon in this discussion.

Grease Lubricated Bearings.

The bearing seal presents perhaps the simplest method of solution with this type of lubrication. As shown in the slides illustrating the two papers, a very effective seal is formed by three grease grooves, the two inner ones which are provided with back drains into the housing.

Felt packing rings are also frequently employed. In order to obtain satisfactory service from felt packings, only high grade soft wool felt should be used and the fit around the shaft should not be too tight to prevent heating in service.

The felt rings should be inspected at the time of the six-month lubrication period and if they have become hard or glazed, they should be taken out and replaced with fresh felt washers.

In order to make this change possible, the felt rings should be split. This split in the felt ring should be angular to prevent leakage, and after placing the ring around the shaft, the ends should be sewn together.

While felt rings of the proper size and properly installed will generally give satisfactory service, there is a tendency for them to absorb the oil contained in the grease, especially at higher temperatures, with consequent oil leakage.

The writer's experience has been that better satisfaction can be obtained by the employment of composite cork or laminated cork rings. The same type of grooves can be employed as for felt packings.

In this connection the writer wishes to draw attention to the design of the bearing housing, which should facilitate the installation, inspection or removal of packing rings by the employment of detachable glands. This is illustrated in several slides of the two papers read.

The writer's experience with cork packings has been very satisfactory, both in the ability to provide a seal, as well as to the life obtainable with them. Properly designed, they can also be employed in connection with oil lubricated bearings.

Composite cork rings will be found more convenient from the replacement standpoint since they can be inserted in halves with the ends cut out and angle or lap-jointed. The two halves are then glued together.

Oil Lubricated Bearings.

As pointed out above, a properly designed cork packing will prove very satisfactory.

Another type of bearing seal which has proven very satisfactory during many years' service is a labyrinth seal of the piston ring type.

It is possible to design such seals in a very simple and effective manner and to employ standard sizes of piston rings, available through the automobile industry.

The writer will be glad to furnish additional information on the design of these packing rings to any one interested in the lubrication of motor bearings.

T. V. Buckwalter*: I will not attempt to add anything in this discussion to the technical description of ball and roller bearings, which has been very well handled by Messrs. Bell and Tawresy, who have respectively presented their papers on the roller and

*Chief Engineer, Timken Roller Bearing Company, Canton, Ohio.

ball bearings. If I could say anything that would be of interest at this time it would be on the general economic situation.

The company with which I am connected is closely related with the automobile industry and it is the general impression that the automotive industry has enjoyed a higher degree of prosperity than many other branches of manufacture and allied industries, and this brings up the point that industrial conditions are rather unbalanced in that some branches are giving relatively greater value per dollar at present than during the war or before the war, whereas other industries are maintaining their prices more nearly in line with war conditions.

I believe it will be generally recognized that the automobile industry and the two large industries with which it is allied, namely, the tire and the petroleum industries, are giving greater value today than at any time in the past, whereas, on the other hand, we can look at certain industries like the furniture business, leather, and clothing industries, on which prices are relatively high as compared with 1914 and which appear to be in a rather bad way, and we hear many complaints that the automobile is robbing them of their just dues.

Now, some of us might take the stand that the automobile and its allied industries are entitled to the business they are obtaining for the reason that they are giving the public so much more for their dollar and that this unbalanced industrial condition will not be corrected until other industries get in line with those that are deflated and bring down their prices more nearly in keeping with pre-war conditions. The railway transportation machine can be considered in this category, and we do not believe that the railway question will be satisfactorily settled until the government bodies get off the backs of the railway managers and allow them to run their properties on an economical basis; that is the only way they can bring their costs down. It is quite apparent, therefore, that the basis for a general period of prosperity will be laid when all of our large industries are as nearly deflated to the normal basis as is the automobile and its related industries.

I am asking your pardon for introducing the above prefacing remarks on general economics into what has been programmed as a technical bearing discussion, and will ask you to pardon this digression with the remark that the company with which I am associated has for some years realized these changing conditions through which we are progressing and they, as a matter of fact, made a start on the program to prepare for these changing conditions even while the war was on, and since the war have been actively engaged in the intensive study of bearing design and manufacturing methods to enable us to give the mechanical industries and the automobile industry better bearing value as regards accuracy and life at lower cost. As a result of this study, the taper roller bearing is being sold at the lowest price in its history and at the same time the quality has been materially improved and this has resulted in an increase in output so that at present this company is supplying upwards of 50 per cent of the anti-friction bearings used in the automotive industry.

Now this brings us to the point as to whether the electrical industry, with which you gentlemen are allied, has gone the limit in the way of improving operating performance of electric motors and reducing cost to the lowest possible point. I am not a skilled electrical engineer, but I believe there is room for considerable improvement in the way of improving the performance of electric motors without material increase in cost, and the time appears to be ripe for an intensive study of the introduction of the rolling type of bearing in the electric motor. Speaking from an operating man's standpoint, I have been informed that almost three-quarters of the maintenance cost of electric motors is due to failure, in one way or another, of the bearings, having in mind, of course, that the riding of the rotor on the stator with the destruction of windings is, strictly speaking, chargeable to bearing failure.

It is recognized that the application of the rolling type of bearing introduces some peculiar problems in the manufacture of the bearing for the reason that the rotating part of an electric motor is relatively heavy and the slight vibratory motion, such as would be set up by inaccuracies of the bearings, is relatively more severe in, say, a 50-hp. armature weighing 500 pounds, than it would be on an automobile pinion shaft weighing 10 pounds and transmitting the same amount of power. Therefore, the bearing for electric motors must be held to rather closer limits than is generally necessary in the automotive industry, and this factor has been recognized by both speakers this evening and I believe is recognized by others who have given considerable thought to this subject.

The subject of bearing mounting as regards taper bearings was touched on lightly by Mr. Bell and a number of views showing the application of ball and roller bearings were shown by both speakers. It is considered advisable to refer at this time to the drawing showing a typical application of the Timken bearing as representative of the taper roller construction to electric motors.

It will be recognized, first, that the bearings are mounted in pairs, that is, in opposition, and that two bearings function as a unit in the support of a rotating element such as an armature. This mounting can be made in a simple manner by mounting the cones or inner races with a press fit on the armature shafts, the roller and cage assembly is integral with the cone and together form a unit. The cup or outer races can be mounted directly in the motor head, can be bolted up solidly at one end, whereas, at the other end thin shims are introduced between the bearing cap and the housing to provide take-up for any variation in length of the bearing, armature shaft and motor frame. There are a number of advantages arising from this take-up or adjustment. In the first place, it is not necessary to manufacture either the bearing or the motor parts to exceptionally accurate length limits, and in the second place the necessity of extremely accurate limits on diameter of the shaft for housing are eliminated. It does not make any material difference to the cone of the taper roller bearing whether it is press fitted to the armature shaft with a fit of, say, .001, .002 or .003 in. It will be tight enough with any of these fits to function as an integral part of the shaft and this is all that is necessary, and if the tighter fit, which gives a little wider tolerance in manufacture, expands the cone a slight amount, say .001 in., it simply adds to the projected length of the bearing and this is automatically taken up by the assembler in his selection of shims to give the desired free-running condition of the bearing in assembly.

Another important point that we must keep in mind on an adjustable bearing is that a certain amount of wear is going to develop in any form of bearing and the taper principle provides a simple and economical means for taking up this wear, as the removal of a shim enables the adjustable cup to be set up closer to the opposing bearing and develops the full contact of the rolls and the races and renews the alignment of the bearing, and at the same time restores the initial air gap. It must not be assumed that because an adjustment is provided that wear is any more rapid than in other types of bearings and it would be interesting in this connection to cite a test that we have just taken down in which the pinion shaft bearings on rear axle, removed after 25,000 miles operation under load, showed a total wear-end-wise of .004 in., or much less than .001 in. radially.

The general subject of lubrication and enclosures for bearings has been very well covered in the previous papers, and in connection with this subject we might mention that we took up the application of the taper roller bearing to electric motors immediately after the war, and one of the principles on which we were working was that the motor must run at full load continuous duty, practically 24 hours per day, for six months without attention or without renewal of the lubricant. Our earlier experiments based on the use of felt enclosure did not produce this result. These early experiments also developed that the bearing must be held in rather close limits as regards accuracy and the continuation of these experiments developed the fact that the spiral annual groove as indicated on the bearing mounting above referred to, in connection with the use of a paste oil or medium grease, produces an enclosure which, in our own experience, has operated for over one year without attention on an overloaded motor running 22 hours per day. This experience on enclosure and lubrication has been duplicated on numerous other motors of various sizes from ¼ hp. up. The spiral groove is made with both right and left spirals to provide for rotation in either direction without change in the mounting. The proper oil or grease level for electric motors is for the enclosure to be about one-fourth to one-third completely filled. This will bring the lubricant level slightly below the bottom of the armature shaft as indicated in the layout.

It has been brought out that it is practically impossible to keep grit and dirt out of the bearing and it should be recognized that dirt and grit will find their entrance in a bearing in many ways, but generally either through the lubricant, through careless assembly and handling, or through the atmosphere. It is also an unfortunate circumstance that the dust in the air and ordinary grit and sand are a little bit harder than hard steel and exercises a lapping action on the rolling type of bearing. The life of the bearing, therefore, is almost entirely dependent on cleanliness as regards the mounting and as regards the maintenance problem being handled in a cleanly manner. As an instance of the absence of wear on a well-maintained electric motor bearing, would state that on the 60-hp. motor above referred to operating for 22 hours a day under overload for over a year, the wear endwise was less than .001 in., and the wear radially about one-fifth of this amount.

The problem of making a bearing quiet is largely one of accuracy in manufacture of the rolling elements, and where these rolls are held down to rather narrow limits as regards roundness, the noise will be very slight and will be practically negligible except possibly on some special applications such as the ventilating ducts of public buildings. The taper bearing is available now with a special lapped roller, with a very high degree of accuracy, and in addition to providing the quiet running so much desired, the rate of wear of the highly finished bearing is very greatly reduced.

There is another factor in connection with the application of the rolling type of bearing to electric motors that has not been stressed to any material extent this evening, and that is over-all length of the motor is materially reduced with the anti-friction bearing. This, we believe, will be recognized by you gentlemen connected with the steel industry.

It should also be mentioned in closing, and which we believe is a point that is not generally recognized, and that is in regard to the extent of the bearing industry. The company with which I am connected is manufacturing upwards of 100,000 bearings per day, which, to the best of our information, represents about one-half of the anti-friction bearing requirements, so that it will be appreciated that an industry of fairly large proportions has been developed around the rolling bearing principle.

R. Pruger*: Some very interesting facts were brought out here this evening by the lecturers and various speakers which recall some points touched upon in a recent meeting, when the importance of correct ball or roller bearing mounting was mentioned. I do not think it necessary to go into this again, but would like to say a few words regarding bearing design factors and the closer getting together of the motor and anti-friction bearing manufacturers.

We have attempted and, I believe, succeeded in establishing a line of standard anti-friction bearings for a certain range of electric motors. While this is an important step in advance, it nevertheless leaves considerable ground which will yet have to be covered. I am trying to bring out the point that when replacement of a plain bearing by an anti-friction bearing has to be made, similar design factors will have to be established for the proper selection of size and type of anti-friction bearing as is now known for the correctly proportioned sleeve bearing. These design factors must consider application, such as belted, geared, chain driven, or direct connected; whether continuous or intermittent service is required; whether severe vibrations and high peak loads may have to be taken care of, and numerous other facts.

In this respect we do not wish to confine ourselves to electric motors, but will have to consider bearings in general and in all lines of industries. That the anti-friction bearings have made their appearance in nearly every line of industry may well be admitted. That being the case, it becomes obvious that design factors must be established which are based upon practical and theoretical facts as well. At the present it is admitted that we have not yet arrived at that stage; however, we are progressing, and it is gratifying to know that ball and roller bearing engineers are now in full co-operation with motor engineers, and are making every effort to settle these questions.

*Mechanical Engineer, Westinghouse Electric & Manufacturing Company, East Pittsburgh, Pa.

S. Farrell*: I do not know of any example to cite regarding the question of heat, with the exception of that slide which was shown, where the ingot is carried from the pits to the approach table, and, of course, that ingot is at the rolling temperature of the steel. Now the trouble they had there was not one with heat conditions, but with the lubricant chamber, which on mill type motor is divided into top and bottom half of the motor. You gentlemen have probably noticed in your tables there is a tendency for oil to leak between the top and bottom half, and that not only goes into the motor winding but goes out, and there is a tendency in operating practice for the man who oils the motors to leave the top cover off of the bearing. In turn the middle ring carries the lubricant back onto the armature shaft. As these gentlemen have mentioned, this was one of the troubles brought about by sleeve bearings. With adoption of anti-friction bearings, it is necessary to confine your arrangement to having that seal on either end, so that regardless of condition of top and bottom of motor, you still maintain them in the wanted position. As to the kind of lubricant, one man recommends one kind and another man another kind. On these mill table drives and screw-downs they use a grease similar to 600-W. I think the trouble is overcome in use by finding a lubricant that will stay in its proper place. There is a tendency for the housing to pound itself on the bearing side. We have furnished numbers of housings to operating men and in their requisitions they have requested over-size housings, which indicates them supporting the bearings, and that gives them an opportunity either to machine that housing uniformly and put in an over-size housing or build it up electrically and machine it to standard. Some go to one extreme and some go back to standard and interchange a bearing or babbit lining with anti-friction. We have a housing which is interchangeable with babbitt lining.

One gentleman asked about changing bearings in the field. I do not think, ordinarily, it is the practice of many operating engineers to keep bearings standard. Sometimes when you lose a bearing in the mill, there is a tendency for some other things to happen when that bearing is loose. For instance, you have a motor in operation for some time; it is good practice to change your armature and bearing attachments, and that gives your shop men an opportunity to look over the armature. He inspects and tightens parts and increases the life of the armature by doing that. But there are times when armatures are so located so you can't change. That is a question that has to be decided by each operator.

J. T. R. Bell: The committee of ball bearing manufacturers and motor manufacturers made certain recommendations. They are supposed to be designing engineers.

Regarding the question of temperatures, we have made some unusual temperature runs up as high as 260 deg.

*Rollway Bearing Company, Syracuse, N. Y.

INSPECTION TRIP

(Continued from page 207)

Electric Department Crane Repair Shop

1—General Electric Company semi-automatic arc welding machine for building up line shaft journals, track wheel treads, etc.

Open Hearth Department

Tilting Furnace Section—
- 3—200-ton tilting furnaces, motor operated.
- 2—150-ton pit ladle cranes.
- 1—75-ton floor crane.
- 2—5-ton charging machines.
- 1—Repair trolley over cranes.
- 3—500-hp. waste heat boilers, 100 hp., a.c. motors on the fans.

Converter and Mixing Section—
- 3—20-ton Bessemer converters, motor operated.
- 1—900-ton mixer, motor operated.
- 1—300-ton mixer, motor operated.
- 1—75-ton ladle crane.
- 1—25-ton crane for dumping iron ladles in mixers.
- 1—5-ton scrap crane.

Stationary Furnaces—
- 6—75-ton stationary furnaces.
- 2—125-ton pit ladle cranes.
- 2—60-ton floor cranes.
- 2—5-ton charging machines.
- 2—5-ton repair trolleys.

Doors of both the tilting and stationary furnaces are electrically operated.
- 1—10-ton Gantry crane for handling scrap.
- 3—10-ton stock yard cranes.
- 1—10-ton semi-Gantry mould yard crane.
- 1—Gas producer coal handling crane, 5-ton capacity.

14-inch and 16-inch Mill

1—400-hp. wound rotor induction motor, 2200 volts, on the roughing rolls.
1—600-hp. wound rotor induction motor with Scherbius speed regulating set, on the finishing stand.
1—10-ton crane in shipping department.

No. 1 Forge Department

1—2000-ton steam press.
1—1200-ton steam press.
1—800-ton steam press.
1—500-ton steam press.
1—300-ton steam press.
1—2500-lb. steam hammer.
1—5000-lb. steam hammer.

Crane Equipment—
- 1—75-ton crane.
- 1—50-ton crane.
- 1—30-ton crane.
- 1—30-ton semi-Gantry crane.
- 1—20-ton crane.
- 1—10-ton crane.
- 2—5-ton cranes.

All of these cranes are operated from platform on the ground close to the press with which the crane works.

Gas Blowing Engine Building

These engines pump air for the blast furnaces.
- 4—Bethlehem Steel Company gas engines.
- 2—Allis-Chalmers gas engines.
- 3—Mesta Machine Company gas engines.
- 1—35-ton crane.
- 1—150-hp. automatically operated air compressors, 250 lbs. pressure.

The ignition for these engines is furnished by small motor generator sets at No. 4 Power Station with a storage battery for emergency.

The gas used by these engines is washed by four Theisen gas washers, each washer being driven by a 150-hp. induction motor.

No. 4 Power Plant

1—3500-kw. 6600-volt, gas driven generator.
2—750-kw. rotary converters, 250 volts d.c.
3—1000-kw. transformers, 6600 volts to 11,000 volts.
3—500-kw. transformers, 6600 volts to 440 volts.
1—250-volt storage battery for gas engine ignition and for operating oil switches and signal system.
1—40-ton crane over power plant.

No. 3 Power Plant

1—2000-kw. steam turbine, 2200 volts, connected through 2200 to 6600 volt transformer, direct to No. 4 Power Plant bus and operated from No. 4 Power Plant.

1—Steam blowing engine for Bessemer converter.
1—Steam blowing engine for the blast furnace.
1—Repair crane over these machines.

Pig Iron Casting Machines

2—35-ton trolleys for tilting the ladles.
2—60-hp. d.c. motors operate the pig iron machines.

Ladle Cleaning House

1—40-ton crane.

Blast Furnace Department

3—500-ton stacks and stoves.
3—250-ton stacks and stoves.

The 500-ton stacks are equipped with Lidgerwood skip hoists and McKee revolving tops. The skip hoists are operated with 150-hp., 250-volt, d.c. motors.

2—7½-ton Alliance ore bridges.
2—50-ton McKee ore transfer cars.
6—Motor-driven scale cars.

Merchant Mill

No. 1 Sub-Station—
1—1000-kw. frequency changer, 2200 volts, 60 cycles, to 440 volts, 25 cycles, used on line from York Haven Water & Power Company.
2—200-kw. rotary converters, 250 volts d.c.
6—200-kva. transformers, 11000 volts to 440 volts.
3—400-kva. transformers, 2200 volts to 440 volts.

20-inch Mill and Splice Bar Shop—
Steam driven 3-high mill; all auxiliary equipment, including tilting tables, is motor operated.
1—35-ton crane over rolls.
1—10-ton crane at furnaces.
1—10-ton stock yard Gantry crane.
1—10-ton crane in splice bar shop.
3—10-ton cranes in shipping yard.
1—480-hp. synchronous motor on air compressor.

Steel Foundry

2—25-ton open hearth furnaces.
1—6-ton Heroult electric furnace.
2—50-ton cranes.
3—25-ton cranes.
2—20-ton cranes.
2—10-ton cranes.
3—5-ton wall cranes.
3—Electric welding machines.

Frog and Switch Department

1—20-ton crane.
2—10-ton cranes.
11—5-ton cranes.
3—3-ton cranes.
27—Reversing motor-driven planers.

Sub-Station No. 11—
3—300-kva. transformers, 11,000 volts to 440 volts.
1—300-kw. engine-driven d.c. generator, 250 volts.
1—300-kw. rotary converter off the 440 volt bus.
1—150-kw. rotary converter off the 440 volt bus.
1—300-kw. rotary converter and transformers connected direct to 11,000 volt lines.

Coke Ovens

120—Semet-Solvay by-product ovens.
60—Koppers by-product ovens.
2—35-ton electric locomotives.
4—75-ton electric locomotives.
3—Oven pushers.
3—Charging cars.
1—Coal bridge for storing coal.
2—150-hp. synchronous motors on gas boosters.
1—Cutler-Hammer Thomas gas meter.

Power Plant No. 5—
1—1500-kw. steam turbine, 440 volts.
2—300-kw. engine generators, 250 volts, d.c.
3—250-kva. transformers, 11,000 volts to 440 volts.

East End Pumping Plant

This plant pumps all the water used by the Steelton plant from the river into the old canal, which acts as a reservoir.

3—125-kw. transformers, 11,000 volts to 440 volts.
3—100-hp. 440-volt vertical motors on pumps.
1—250-hp. 440-volt synchronous motor on pump.

Bridge and Construction Department

1—65-ton Gantry crane.
2—50-ton Gantry cranes.
2—25-ton Gantry cranes.
2—50-ton cranes.
1—25-ton crane.
9—15-ton cranes.
2—10-ton cranes.
7—10-ton wall cranes.
13—5-ton wall cranes.
30—3-ton wall cranes.

No. 21 Sub-Station—
6—Transformers, 11,000 volts to 440 volts, 300 kva.
2—300-kw. rotary converters, 250 volts, d.c.
2—Motor driven air compressors, 300-hp. motors.
1—Motor driven air compressor, 50-hp. motor.
1—Steam driven air compressor.

General Data

Total number of cranes on plant	220
Total number of motors on plant	2,500
Total horsepower of motors on plant	58,000
Kwh. generated per month	6,250,000
Kwh. purchased per month	400,000

Have you read over pages 202 and 203 of this issue?

They contain floor plan and application blank for space at our Iron and Steel Exposition to be held September 15, 16, 17, 18, 19 and 20, at Duquesne Garden, Pittsburgh, Pa.

JUNE ISSUE

of

Iron and Steel Engineer

will contain all the papers and discussions presented before the Combustion Engineering Section of the A. I. & S. E. E. at their Fuel Savings Conference held April 2nd and 3rd at the William Penn Hotel, Pittsburgh, Pa.

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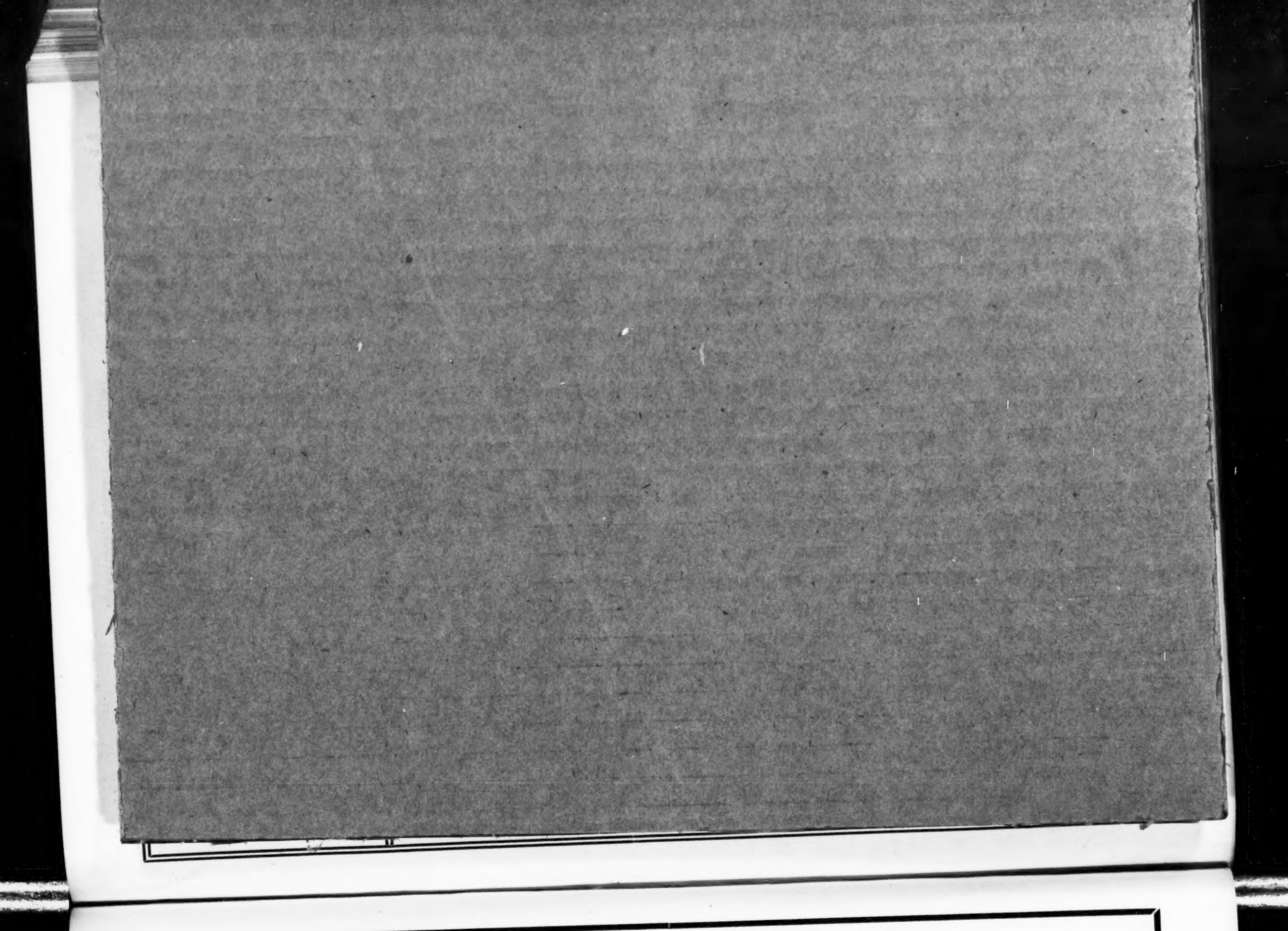
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